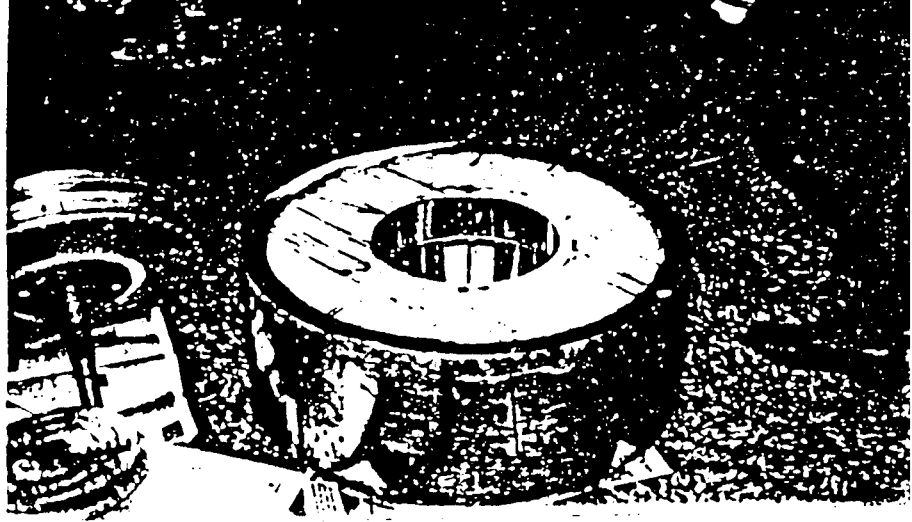


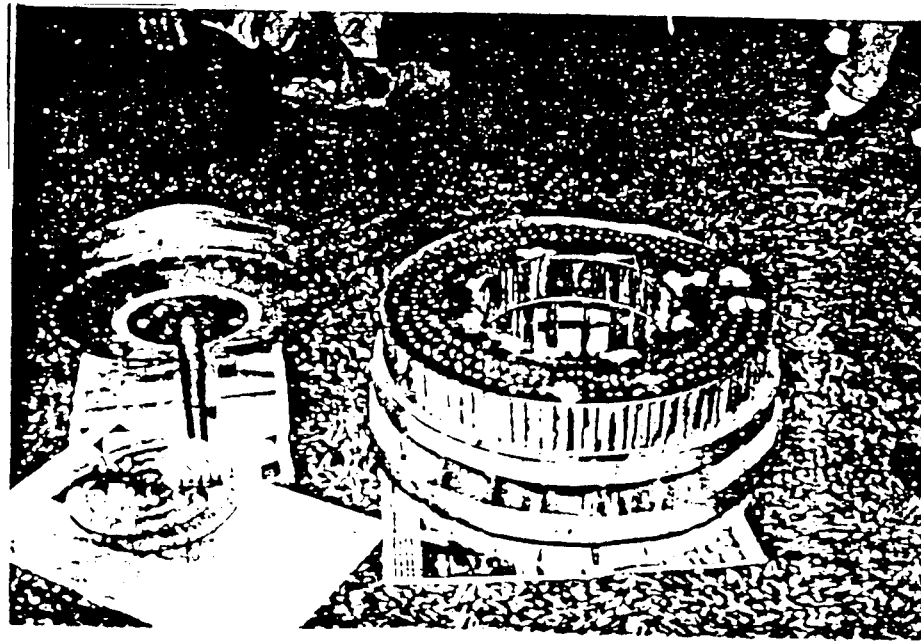
ESKELI'S WORK-FREE HEAT PUMP



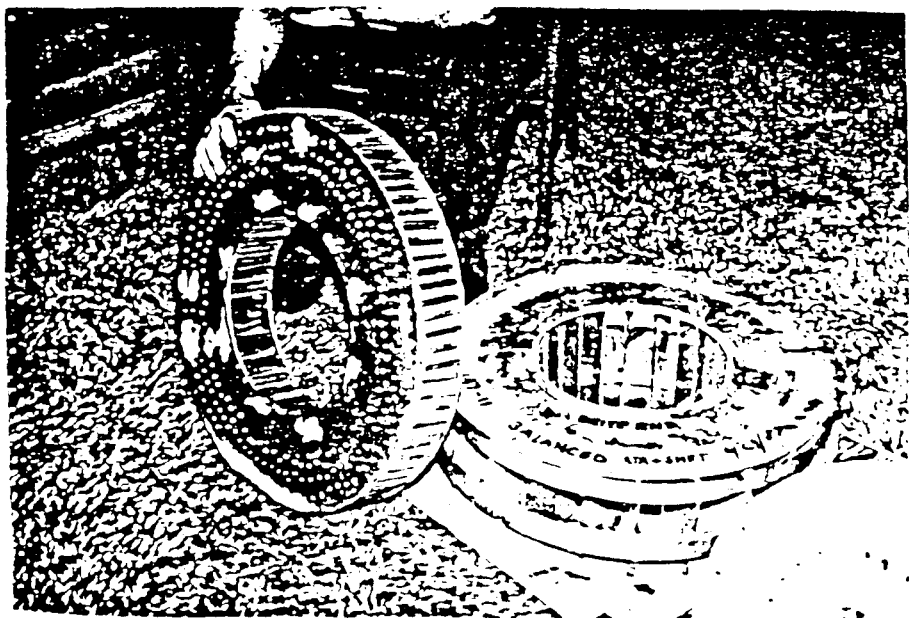
Engine with Housing ON

Mike Eskeli

His Gas Turbine uses compressed Argon or air as a working fluid. When the gas is pressurized and the turbine is rotated by an outside source of motor (electric motor), the gas is compressed to generate heat. As the working fluid moves through the high to low pressure zones, the engine begins to rotate using the sole power of the flow of hot to cold which is partially supplied by outside sources of low temp heat. This heat is "entrained" within the engine to be pumped, to a higher temperature which can provide energy for practical work.



Engine with Housing OFF, Turbines exposed



High speed turbine separated from low speed turbine

The Plenum. Vol. 1, no. 1,
July/August 1989.
(Vanguard Sciences).

The WORK-FREE HEAT PUMP

by M. Eskeli, Houston.

Heat pumps are well known, and are used for heating and cooling applications. In the usual heat pump, the working fluid is compressed and condensed, and then expanded and evaporated, and the process requires large amounts of work, which is the "price" required for the transfer of heat from a lower temperature to a higher temperature by the machine.

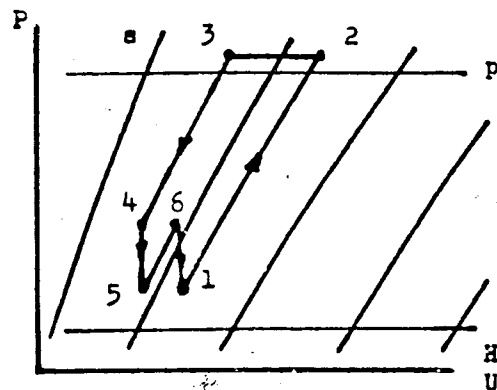
A new heat pump work cycle, and the equipment to perform that cycle has now been developed. The work cycle is shown below, and is applicable for all types of equipment.

THE WORK CYCLE

The work cycle is shown at right and the heat pump cycle is 1-2-3-4-5-6-1. 1-2, 3-4 and 5-6 are isentropic processes 4-5, 6-1 are at approx. constant temperature, and 2-3 is a constant pressure heat removal process.

Working fluids typically are gases such as nitrogen, N_2 ; oxygen, CO , and others.

The work process may be either steady flow, or it may be a non-flow process.



Pressure-enthalpy, or
pressure-internal energy diagram.

For suitable gases, including those listed, and for suitable temperatures, the work cycle is WORK FREE, so that heat is pumped from temperatures represented by 4-5 and 6-1, into temperatures represented by line 2-3. The heat is pumped without work to the machine, for the heat transfer process.

Brief Example: Use nitrogen, and set values at noted points as follows:

point	1	2	3	4	5	6
press.	150	540	540	250	150	250
						psia.
Temp.	60	280	138	+4	+4	60
						°F

(Additional values for the gas can be obtained from gas tables)

Work: 1-2=53.2	Total for compression = 66.7 B/lb	Work: 3-4=31.6	6-1=18.7
5-6=13.5	Total for expansion = 67.0 B/lb	4-5=16.6	

Thus, for the closed loop of the cycle, work input is nil.

Heat is delivered 2-3= 36.8 B/lb; between temperatures of 280F and 138F

Heat is supplied to the heat pump 4-5 and 6-1, total 37 B/lb. (calculated) at 4F and 60F

Thus, the calculation indicates that the cycle is in balance, approximately.

The above brief example was based on a steady flow type apparatus, and enthalpy values were used. Even better results can be obtained, where a non-flow process is used.

Variety of equipment can be used to pump heat using the work cycle shown. One type of equipment, of relatively simple design, is shown on opposite side of this sheet.

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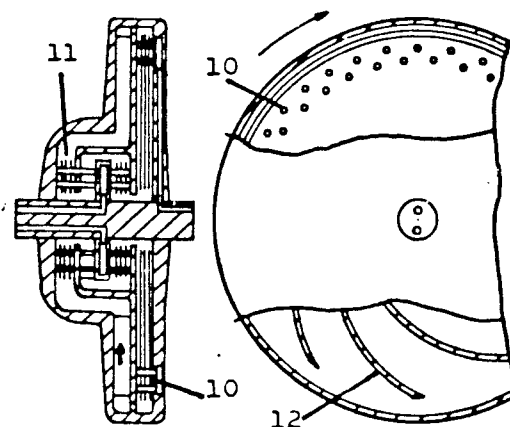
THE CENTRIFUGE TYPE HEAT PUMP.

The work cycle shown on opposite side of this sheet can be readily performed by the centrifuge type heat pump. This is a unit which has only one moving part, the rotor, and the working fluid, such as nitrogen, is sealed within the rotor, and circulates therein within passages of the rotor. The circulation of the working fluid within the rotor is accomplished by density control alone, in accordance with the work cycle shown, and there is no work input to the working fluid from the rotor shaft. Thus the work input for the heat transfer is nil, and a WORK-FREE heat pump results.

In the diagram-drawing at right, an axial cross section and an end view with sections removed shows a typical heat pump rotor suitable for use with the work cycle shown on the other side of this sheet.

10 is heated fluid heat exchanger, 11 is heat supply heat exchanger in two parts, and 12 are vanes in passages that return the working fluid from periphery to center.

The work cycle process is of the non-flow type in this rotor for the working fluid.



Centrifuge type heat pump.

The fluid to be heated is usually a liquid, such as water, and enters and leaves the rotor via rotor shaft. Similarly, the heat supply fluid circulating through heat exchanger 11, is a liquid and enters and leaves via rotor shaft passages.

APPLICATIONS

The WORK FREE HEAT PUMP has obviously many uses. One such use is in heating of all types of buildings and homes, resulting in cost free heating, since no fuel is needed, and the power usage is nearly nil. (In the above shown heat pump, power is needed to drive the rotor against friction, which may require a $\frac{1}{4}$ - $\frac{1}{2}$ hp motor).

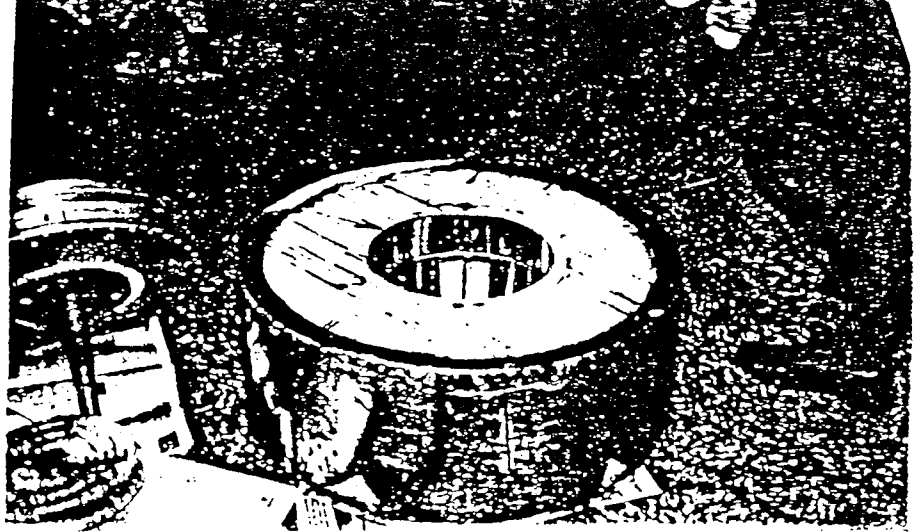
Another use is in power generation, resulting in cost free power, since the unit uses no fuel, the energy source being either ambient air, or water from some natural source. (Attached turbine generates the power; part of this is used to overcome the heat pump friction loss, and remainder is available for generation of electricity.) Further uses are in portable power, automobiles, other transportation, etc.

The apparatus and methods and work cycles are patented, or have patents pending. For basic heat pump, see US. Pat. 3,926,010, and Canadian pat. 984,827. Patents also pending in foreign countries, by M. Eskeli, 7994-41 Locke Lane, Houston Texas 77042; 713-783-8776.

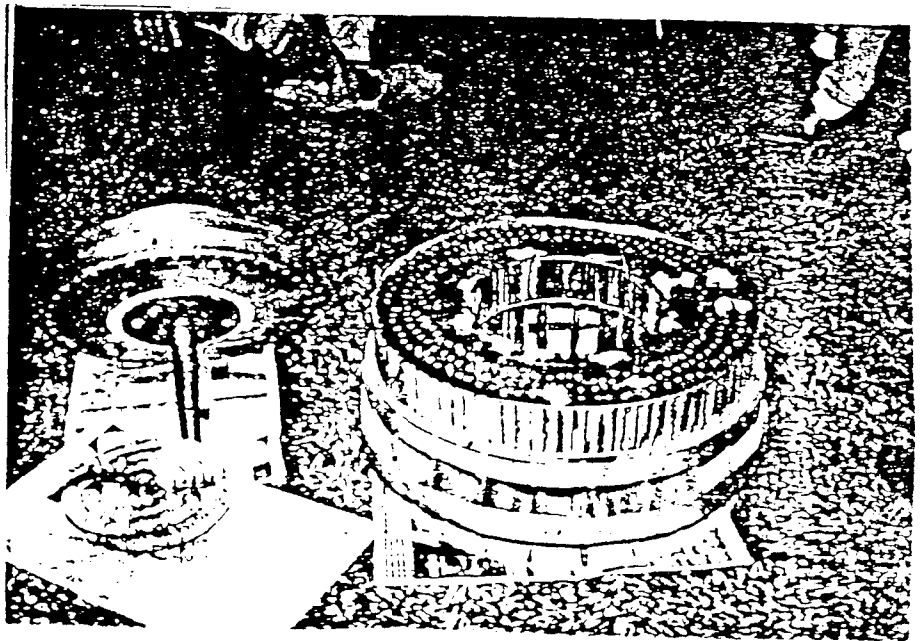


Mike Eskell

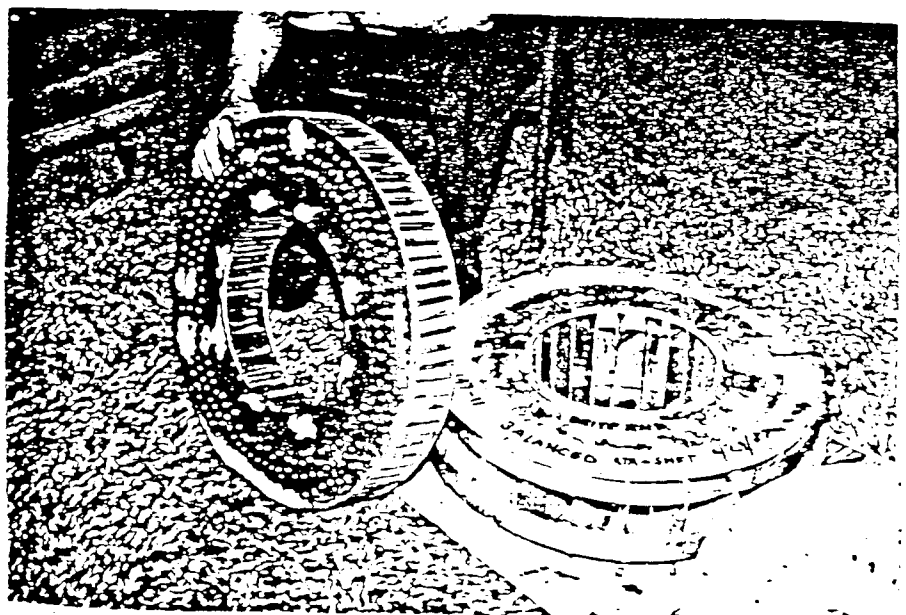
His Gas Turbine uses compressed Argon or air as a working fluid. When the gas is pressurized and the turbine is rotated by an outside source of motor (electric motor), the gas is compressed to generate heat. As the working fluid moves through the high to low pressure zones, the engine begins to rotate using the sole power of the flow of hot to cold which is partially supplied by outside sources of low temp heat. This heat is "entrained" within the engine to be pumped to a higher temperature which can provide energy for practical work.



Engine with Housing ON



Engine with Housing OFF, Turbines exposed



High speed turbine separated from low speed turbine

*The Plenary
Vanguard Science
Vol 1 NO 1
5th Aug 89*

Free Energy Update

Eskeli Thermodynamic Engine

In April of this year, Paul Carlson made us aware of a local inventor, Mr. Michael Eskeli, who had developed two types of engines which take advantage of the temperature differential utilized in Thermodynamics to develop a motive force.

Mr. Eskeli became incensed at an article written in the Dallas Times Herald concerning the failure of science to come up with alternative power systems not dependent on petroleum technologies. His concern prompted a letter to the editor on April 24 89.

"I have had the patents for fuelless power generators, work-free heat pumps and other related items, 56 of them issued in the mid-70s, and am still waiting to collect a first nickel for labors and expense. And the indications are that these machines will not be in use before 2110 or so."

"The same probably will be true for the fusion people, excited by their discoveries now. There simply is too much oil still around to allow competition."

Based on this letter, we contacted Mr. Eskeli and found him to be authentic in his statement that he did indeed have numerous patents on fuelless engines and related items.

We consequently asked him to speak to a meeting of the Vanguard group to explain his principles of operation. Several weeks later, a demonstration was arranged to actually witness the device in action. Mr. Eskeli agreed to allow a very small group to meet with him in his home and agreed to demonstrate his device to a single person. Pictures of his disassembled Gas Turbine engine are included in this newsletter to show the principle of operation.

The meeting took place and Eskeli allowed two of us to witness the engine in his workshop the next day. We found a device with an approximate diameter of 4 feet and a length of 5 feet which was attached to a huge electrical motor by a belt. Michael explained that he had a small problem which did not allow the unit to operate under its own power, so he simply powered it from an electric motor to let us see how it worked. Subsequent communications have been unrewarding in the hope of a "real-time" demonstration. Until a working model is produced, we have decided to press on to other technologies and systems...JWD

United States Patent

Eskeli

[15] 3,650,636

[45] Mar. 21, 1972

[54] ROTARY GAS COMPRESSOR

[72] Inventor: Michael Eskeli, 2932 Sandage Ave., Fort Worth, Tex. 76109

[22] Filed: May 6, 1970

[21] Appl. No.: 35,112

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Primary Examiner—C. J. Husar
Attorney—Wofford, Felsman & Fails

[52] U.S. Cl. 417/78, 415/1, 415/213
[51] Int. Cl. F04b 23/04
[58] Field of Search 415/1, 72; 417/78

[56] References Cited

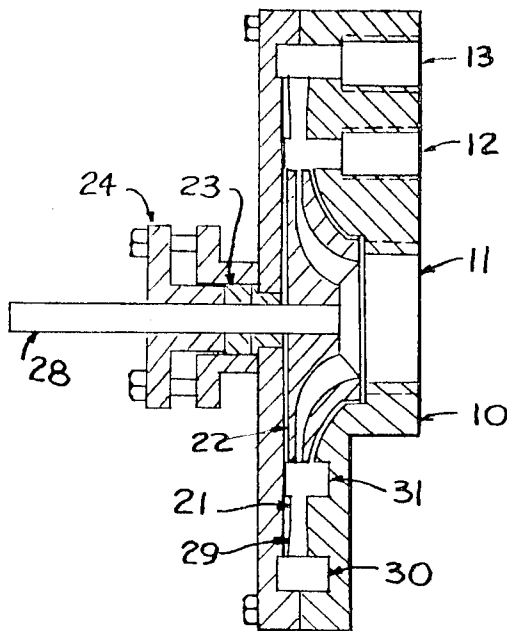
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[57] ABSTRACT

Method and apparatus for a compressor for compressing air, gases and vapors isothermally using a liquid stream to compress the gas; the liquid issuing from an impeller intermittently, with the gas being entrained between these liquid pulses and compressed by the liquid; the liquid having high kinetic energy when leaving the impeller and in slowing the kinetic energy is converted to pressure for both the liquid and entrained gas. Also, this compressor may be used advantageously to compress vapors, wherein the liquid is the same fluid as the gas, in which case condensation of the gas to the liquid occurs, and work of compression is reduced.

2 Claims, 4 Drawing Figures



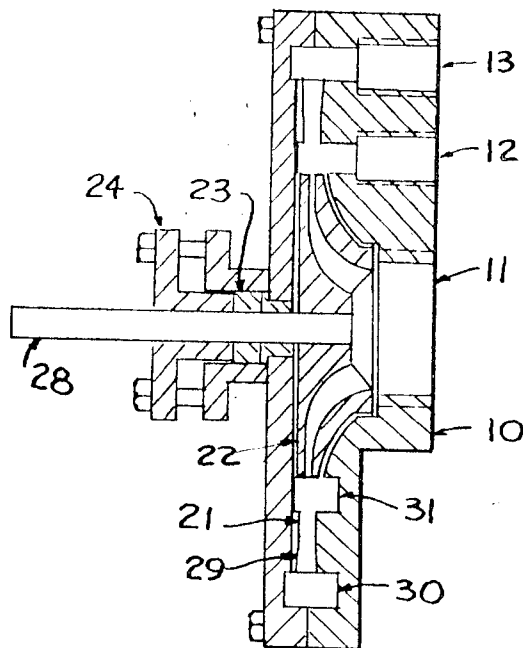
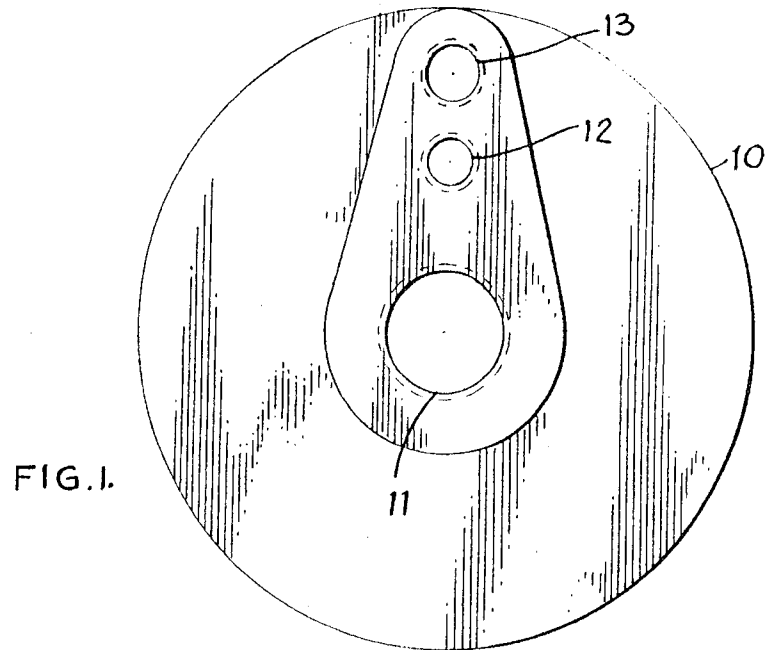


FIG. 2.

INVENTOR.

BY *Michael Eskeli*
MICHAEL ESKELI

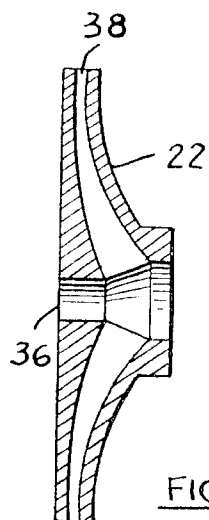


FIG. 3

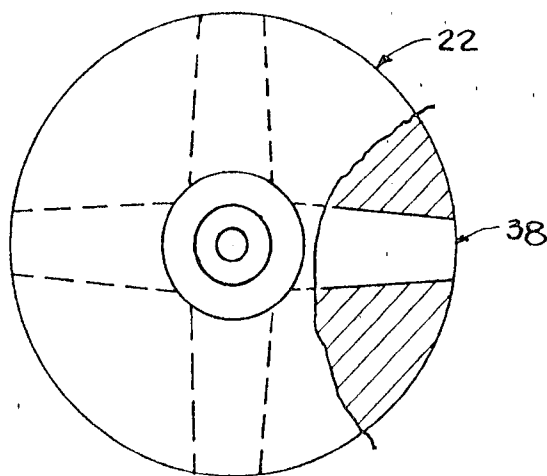


FIG 4

INVENTOR.

BY *Michael Eskeli*
MICHAEL ESKELI

ROTARY GAS COMPRESSOR

BACKGROUND OF THE INVENTION

This invention relates generally to devices for compressing gases, air and vapors, in which a liquid is in intimate contact with the gas or vapor to be compressed.

DESCRIPTION OF PRIOR ART

There are numerous devices and machines available for compressing a gas or a vapor. In some of these machines a liquid is rotated inside an eccentric casing, so that the machine rotor will cause the liquid to pulsate and the space between the rotor blades is increased or decreased, and this variation compresses the gas. These machines are called liquid piston type machines. Another device is the jet ejector compressor, where a stream of liquid or gas is used to entrain the gas or vapor to be compressed, and the kinetic energy of the stream is converted in a diverging nozzle to a pressure.

The main disadvantage of the liquid piston type machine is its poor efficiency, since the liquid is rotated in the machine and requires relatively large power input for compressing the gas. In the ejector compressor, the velocity of the liquid stream is limited and it entrains poorly of any gas; therefore the efficiency of the device is very poor. The available kinetic energy in the liquid stream is high, but due to poor entrainment of the gas by the liquid, results for the device are poor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an end view of the compressor casing, showing the exterior.

FIG. 2 is a side view and a section of the casing and the impeller of the compressor.

FIG. 3 is a side view and a section of the impeller, and

FIG. 4 is an end view of the impeller, showing the fluid passages.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and a device for compressing gases or vapors essentially isothermally in which the kinetic energy contained by a liquid stream is used to compress said gas to a higher pressure where the liquid in slowing in speed will increase its pressure and increase the pressure of the gas being entrained in it. Also, it is an object of this invention to provide a method and a device in which the gas may be partially or fully be condensed in the liquid stream thereby lowering the work of compression; this occurring when the gas or vapor being compressed is the same fluid as the liquid; that is, the gas being compressed is the vapor phase of the fluid, and the liquid being used for as the motive fluid is the liquid phase of the fluid.

Referring to FIG. 1, there is shown an end view of the compressor. 10 is the compressor casing, 11 is the liquid inlet, 12 is the gas or vapor inlet, and 13 is the outlet.

In FIG. 2, a side view of the compressor is shown. The impeller 22 is rotated by shaft 28, supported by bearings and sealed by packing 23 and stuffing box 24. Alternately a mechanical seal could be used. The liquid that is used as the motive fluid enters through opening 11, passes through the impeller 22 and leaves the impeller at a high velocity and entering the throat section 21 and from there the diffuser section 29 in the casing 10. After leaving the diffuser at a higher pressure, and at a lower velocity, the gas and liquid mixture is collected in annular space 30, and from there passes out through opening 13. The liquid entrains gas from annular space 31, and the gas enters the annular space from outside through opening 12.

In FIG. 3, the impeller 22 is shown in more detail. 38 is the fluid passage, and 36 is the opening for the drive shaft.

In FIG. 4, the impeller is shown, with 22 being the impeller and 38 being the fluid passage.

In operation, the compressor functions in a manner similar to a jet ejector compressor. A motive fluid is accelerated in a passage in the impeller to a high velocity; this corresponds to the motive fluid nozzle in a jet ejector. However, the fluid stream issuing from the impeller, when it rotates, is not con-

tinuous as seen by the compressor casing, since in this particular instance, the impeller has four fluid passages, with solid material between. Therefore, the flow from impeller, as seen by the compressor casing, is pulsating, with empty spaces between the high speed liquid; these empty spaces being filled by the gas from the annular spaces, item 31, FIG. 2, and the gas being rapidly moved with the liquid to the outer annular space 30, and from there to discharge. This pulsating action improves the entrainment of the gas by the liquid, and more fully utilize the kinetic energy available in the liquid stream.

The sizing of the fluid passages and the calculations pertaining to same are fully described in thermodynamics literature for jet ejectors and for steam injectors. The space of the passage 38 in FIG. 3, would be either converging for liquids that do not vaporize when leaving the passage; or the passage could be diverging at its outlet for fluids that will vaporize either partially or fully when leaving the passage. Of the non-vaporizing liquids, water would be an example, and of the partially vaporizing types, butane would be an example, both at atmospheric temperatures, and at low pressures. As illustrated in FIGS. 2-4, passageways 38 comprise a converging section nearest the center of the impeller but are at least non-converging at the discharge section. Preferably, the at least non-converging section is a diverging section for better taking advantage of the energy available in the motive fluid to effect higher effluent velocities thereof.

The fluid passages shown in FIG. 4, item 38, can be radial as illustrated, or be forward or backward curved, depending on the fluid used or the shape of the passages. Also, the throat section 21, FIG. 2, may have vanes of proper shape to prevent circular motion of the fluid after it leaves the impeller. Vanes of this type are commonly used in turbines and pumps and are not described herein. Number of fluid passages in FIG. 4 is indicated to be four, but this number would be as required when calculations are made pertaining to the size of the passages, and the frequency of pulses of liquid required to maintain suitable pressure and volume relationships inside the compressor; also, the rotational speed of the impeller would enter into these calculations.

Normally, the amount of liquid as compared to the amount of gas or vapor, is large. Therefore, when compressing a gas, the heat of compression from the gas is transferred to the liquid, resulting in a temperature increase for the liquid, as well as the gas. This temperature increase is much less than it would be for the gas alone, resulting in nearly isothermal compression, and therefore reduced work of compression, as compared to isentropic compression that is often used in rotary compressors. Also, if a liquid that will expand in the impeller is used, with an expanding fluid passage, the temperature of the motive fluid is lowered, and the fluid velocity greatly increased, resulting in much better efficiency for the compressor; this is similar to the function of converging-diverging diverging nozzles in jet ejectors.

The operation of the compressor may be inferred from the above descriptive matter. A liquid source is connected to the impeller inlet, and a gas or vapor source is connected to the gas inlet, FIG. 1, 11 and 12, respectively. Discharge from the compressor is from 13, FIG. 1. A suitable power source, such as an electric motor, is connected to shaft 28, FIG. 2, causing the shaft to rotate. The liquid is accelerated by the action of the impeller, and as it passes through the annular space 31, FIG. 2, in pulsating flow, it entrains the gas and carries it to annular space 30, and from there to discharge.

Materials of construction for the compressor would be similar to those used to make pumps for pumping liquids. Cast iron, steel, bronze, brass, stainless steel and various plastics could be used.

What is claimed new is as follows:

1. A machine for compressing gaseous fluid and having the major components of:

a. an impeller for accelerating a motive fluid to a high velocity; said impeller having a plurality of passageways that comprise respective initially converging sections as

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the passageways extend outwardly from the center of said
impeller and at least non-converging sections exteriorly
of said converging sections; said non-converging sections
defining the discharge passageways of said impeller for
more effective use of the available energy of said motive
fluid which has been accelerated to high velocity whereby
a motive fluid may be partially vaporized at the decreas-
ing pressure due to said high velocity to attain even higher
velocities for more effective entrainment of said gaseous
fluid; and
b. a casing for the compressor, said casing including a dif-
fuser section for slowing the high speed mixture of fluids

and converting the kinetic energy of the stream to pres-
sure, said diffuser section containing a throat section
where the mixing of the motive fluid from the impeller
and the vapor to be compressed occurs, and a plurality of
suitable annular spaces disposed peripherally exteriorly
of said impeller respectively for the entering gaseous fluid
and for the mixture of motive fluid and gaseous fluid and
respective apertures for the entering gaseous fluid and the
effluent mixture of fluids.
2. The machine of claim 1 wherein said at least non-con-
verging section is diverging.

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[54] **ROTARY EJECTOR COMPRESSOR**

[76] Inventor: **Michael Eskeli**, 2932 Sandage Avenue, Fort Worth, Tex. 76109

[22] Filed: **March 30, 1971**

[21] Appl. No.: **129,407**

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Primary Examiner—Henry F. Raduazo

Attorney—Wofford, Felsman & Fails

[52] U.S. Cl. 417/78, 415/98, 415/211, 417/161

[51] Int. Cl. F04b 23/04, F04f 5/00

[58] Field of Search ..415/97, 98, 116, 211; 417/161, 417/78, 116, 211, 98, 97

[56] **References Cited**

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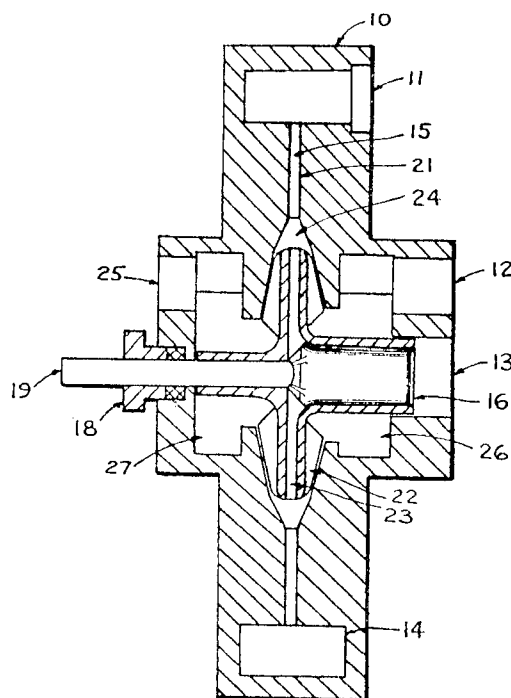
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[57]

ABSTRACT

Method and apparatus for a compressor for compressing air, gases and vapors approximately isothermally by using a liquid stream to entrain and compress the gas; with the liquid being accelerated by an impeller, inner passage, and the gas initially being accelerated by vanes placed on the outside of the same impeller; the two streams are brought together and compressed by using the kinetic energy contained in the liquid stream. Further, the compressor can advantageously be used to compress vapors where the gas stream is condensed fully or in part by the liquid stream thereby decreasing the specific volume of the total fluid stream during compression, and improving the efficiency of the compression.

6 Claims, 3 Drawing Figures



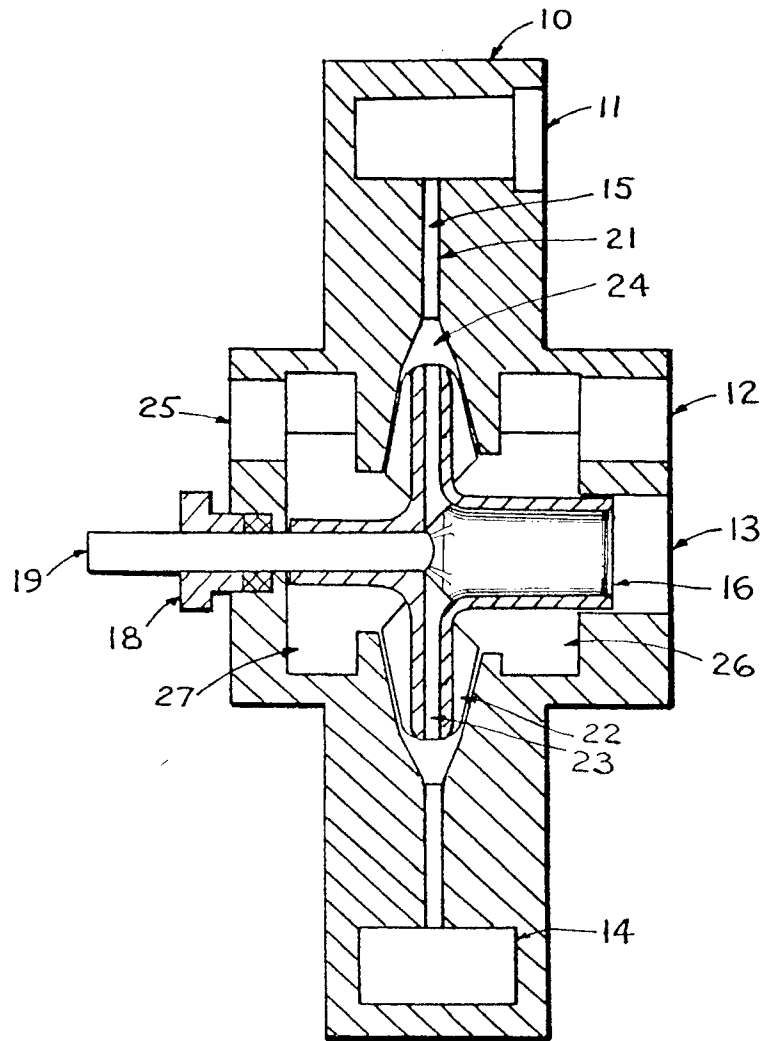


FIG 3

INVENTOR,

BY *Michael Esheli*

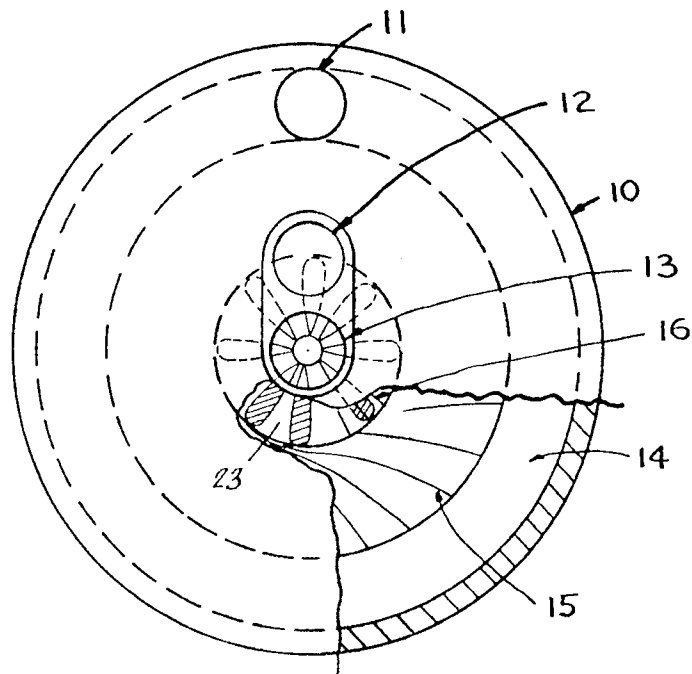


FIG 1

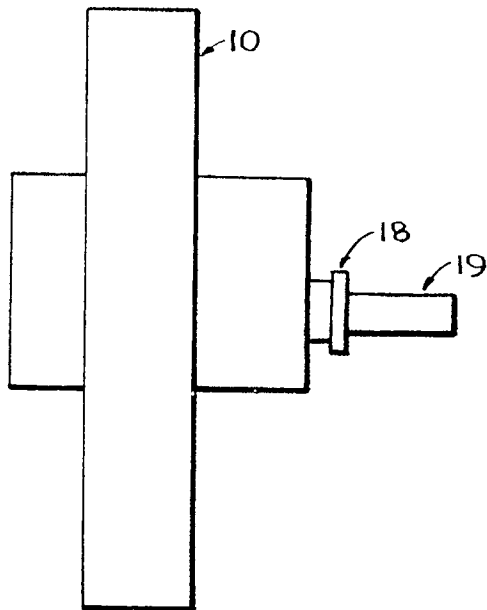


FIG 2

INVENTOR.

BY *Michael Eskeli*

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ROTARY EJECTOR COMPRESSOR

BACKGROUND OF THE INVENTION

This invention relates generally to devices for compressing gases or vapors in which a liquid fluid is in intimate contact with the gas or vapor to be compressed during compression.

DESCRIPTION OF PRIOR ART

There are numerous devices and machines available for compressing a gas or a vapor. In some of these machines a liquid is rotated inside an eccentric casing causing the liquid to pulsate and compress the gas; these are usually referred to as liquid piston compressors. Another device is the jet ejector compressor, where a high velocity fluid stream is used to compress a gas or a vapor.

The main disadvantage of the liquid piston type machine is its poor efficiency due to the liquid being circulated in the machine casing requiring large amounts of power. In the ejector compressor, when liquid is used as the motive fluid, the velocity of the liquid stream is limited in the motive fluid nozzle by the nature of the device, and further entrainment of the gas by the liquid is poor in many instances, reducing the efficiency of the device.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an exterior view of the compressor casing, and

FIG. 2 is a side exterior view of the compressor casing.

FIG. 3 is a cross section of the compressor illustrating the interior construction details of the compressor.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and a device for compressing gases or vapors in which the kinetic energy contained in a fluid stream is used to compress said gas or vapor to a higher pressure. Also, it is an object of this invention to provide a method and a device where the gas or vapor to be compressed may be either partly or fully be condensed in the high velocity fluid stream during compression thereby decreasing the required work of compression.

Referring to FIG. 1, there is shown an end view of the compressor. 10 is the compressor casing, 11 is the fluid outlet from the compressor, 12 is the gas or vapor inlet to the compressor, 13 is the liquid motive fluid inlet to the impeller, 14 is an annular space to which the compressed fluid is collected, 15 is one of the vanes in the compressing passage, 16 is the impeller with its vanes for the initial acceleration of the gaseous fluid.

In FIG. 2, a side view of the compressor is shown. 10 is the compressor casing, 18 is the shaft seal, and 19 is the impeller shaft.

In FIG. 3, a cross section of the compressor is shown. 10 is the compressor casing, 11 is the outlet for the compressed fluid, 12 and 25 are the two inlets for the suction fluid, 13 is the liquid motive fluid inlet, 14 is the annular space for collecting the fluid after compression, 16 is the impeller, 18 is the shaft seal, 19 is the impeller shaft, 21 is the passage where the fluid is compressed and the kinetic energy contained in the fluid mixture is converted to pressure, 15 is the vane in the

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passage 21, 24 is the mixing area where the two fluid streams are brought together, 22 indicates the vanes on the exterior of the impeller used to initially accelerate the gaseous portion of the fluid to be compressed, 23 indicates the passage for the liquid portion of the fluid with suitable vanes, and 26 and 27 are two spaces for passing through the gaseous portion of the fluid.

The function of the compressor is as follows. Liquid fluid is passed to the center of impeller, 16 in FIG. 3. The liquid is accelerated in the impeller by centrifugal action, and by a pressure differential, since the liquid fluid normally is coming from the receiver-separator where the pressure is the same as at the outlet of the compressor. Also, with certain fluids, where the fluid in the reservoir is a saturated liquid with saturated vapor in the upper part of the said reservoir, the liquid fluid in the impeller is partially vaporized and this vaporization further accelerates said fluid in the impeller passage 23, FIG. 3. The gaseous or vapor fluid entering the compressor through openings 12 and 25, is passed to the external vanes of the impeller and there accelerated; this acceleration will improve entrainment of the gas in the motive fluid in the mixing area 24 and also provide some initial velocity for the gas and thus decrease the shock in the motive fluid stream which a stationary gas would impose. After mixing, the fluid stream is compressed in the passage 21, with a corresponding decrease in velocity. After collection of the fluid in space 14, the fluid is passed to a reservoir-separator through opening 11.

The passages in the impeller, 23, must be so designed that the rotational energy available from the impeller, and the pressure energy available in the liquid, and the energy available due to possible gasification of the liquid in the passage, are all converted to a velocity energy at the passage outlet.

As illustrated in FIGS. 1 and 3, the passages 23 comprise a converging section nearest the center of the impeller but are at least non-converging at the discharge section. Preferably, the at least non-converging section is a diverging section for better taking advantage of the energy available in a saturated liquid motive fluid to effect higher effluent velocities thereof at a substantially continuous peripheral discharge passage of the impeller. Similarly, the passage in the stationary casing, 21, must be so designed that the kinetic energy in the fluid stream is converted to pressure. As also illustrated in FIGS. 1 and 3, the passageways 21 intermediate the vanes 15 are similarly at least non-converging, and preferably diverging. These calculations are not detailed herein, since they are available in various textbooks on fluid mechanics and on jet ejector compressors.

The compressor may be used to compress gases, such as air, by using a liquid, such as water, as the motive fluid. Alternately, a vapor, such as propane, may be compressed using as the motive fluid propane liquid. Normally, the amount of liquid as compared to gas, is large, and in the case, such as air, the compression is approximately isothermal, resulting in a lower work of compression. Where the same fluid is used both as the motive and the suction fluid, liquid and vapor respectively, the vapor is partially condensed in the liquid, resulting in a lower work of compression and improved efficiency.

The operation of the compressor may be inferred from the above descriptive matter. A liquid source is connected to the impeller inlet, and a gas or vapor source connected to the gas inlet. Discharge from the compressor is through the discharge opening. A suitable power source, such as an electric motor, is connected to the compressor shaft, causing it to rotate in suitable direction. The liquid is accelerated by the impeller, by the pressure differentials within the impeller and by possible vaporization of the liquid fluid. After leaving the impeller, the gaseous fluid and the motive fluid are mixed, and then compressed in the stationary part of the compressor, and then discharged from the compressor.

Further, the said compressor may be built into a pressure vessel or a tank, with the impeller liquid suction from such tank directly. In such arrangement, the annular space 14, FIG. 3, may be omitted, with the compressor discharging directly to the said tank. The gas suction 25, FIG. 3, may open through the tank wall to outside and a suitable conduit connected through the tank wall to opening 12.

What is claimed is:

1. A machine for compressing gaseous fluid and comprising:

a. an impeller for accelerating a liquid motive fluid to a high velocity; said motive fluid comprising at the entry to the impeller a liquid; said impeller having fluid passages that comprise respective initially converging sections as the passages extend outwardly from the center of said impeller and at least non-converging sections extending radially exteriorly of said converging sections; said plurality of non-converging sections defining a substantially continuous discharge passage of said impeller for continuous flow of said motive fluid therefrom and for more effective use of the available energy of said motive fluid which has been accelerated to a high velocity such that said motive fluid may be partially vaporized at the decreasing pressure due to said at least non-converging passages and said high velocity to attain even higher velocities for more effective entrainment of said gaseous fluid; and being designed to convert to velocity of said motive fluid, energy due to centrifugal action of the rotating impeller and also energy due to pressure differential between the impeller inlet and the impeller outlet;

b. a casing disposed about said impeller; said casing having a centrally disposed chamber defined about said impeller for receiving said gaseous fluid; said

chamber extending radially outwardly concurrently with said impeller; an inlet port connected with said chamber for intake of said gaseous fluid; a mixing section connected with the circumferential portion of said chamber and encompassing the discharge passage of said impeller for entraining said gaseous fluid from said chamber into the effluent stream of said motive fluid which has unusually high velocity and unusually high efficiency of entrainment as a consequence of its acceleration by said impeller; said casing having a suitable compressing passage providing for the compression and deceleration of the combined fluid stream after said motive fluid leaves said impeller, suitable vanes being provided in said compressing passage to prevent circular motion of said fluid in said compressing passage and to improve compression performance; said casing being provided with inlet and outlet ports, respectively, for connection of suitable respective conduits for the intake of said motive fluid and for the discharge of said compressed gaseous fluid and said motive fluid; and

c. a shaft to support said impeller, said shaft having a shaft seal.

2. The machine of claim 1 wherein the impeller motive fluid passages are diverging to provide for acceleration of the motive fluid due to partial vaporization of said motive fluid in said impeller fluid passages.

3. The machine of claim 1 wherein the mixing section and the compressing passages in the casing containing the vanes, are made a converging-diverging passage with the position of the vanes being adjustable to provide for optimum performance.

4. The machine of claim 1 wherein said machine is used as a vacuum pump.

5. The machine of claim 1 wherein gas acceleration vanes are provided on the outer surface of said impeller intermediate said impeller and the interior walls of said chamber for accelerating said gaseous fluid and for providing an initial velocity to said gaseous fluid so as to move it in the same direction that said motive fluid is moving such that the entrainment of said gaseous fluid is enhanced for improved efficiency in compressing said gaseous fluid by said high velocity motive fluid from said impeller.

6. The machine of claim 1 wherein the compressor is built into a pressure vessel, with the impeller suction being from said vessel, and the compressor discharge being directly to said pressure vessel.

* * * * *

[54] REACTION TURBINE

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[22] Filed: Nov. 1, 1971

[21] Appl. No.: 194,205

[57] ABSTRACT

[52] U.S. Cl. 415/80

[51] Int. Cl. F01d 1/18

[58] Field of Search 415/80

A method and apparatus for generating power in response of a fluid flowing from a higher pressure to a lower pressure through a reaction turbine. Said fluid enters said turbine rotor wheel near the center, is then pressurized within said rotor wheel and then discharged via exit nozzles mounted on said wheel in a backward direction; said fluid discharge generating a torque on said wheel, said torque then being passed to the rotor shaft as the useful work output of said turbine. Fluids used may either gaseous, liquids or mixtures of the two.

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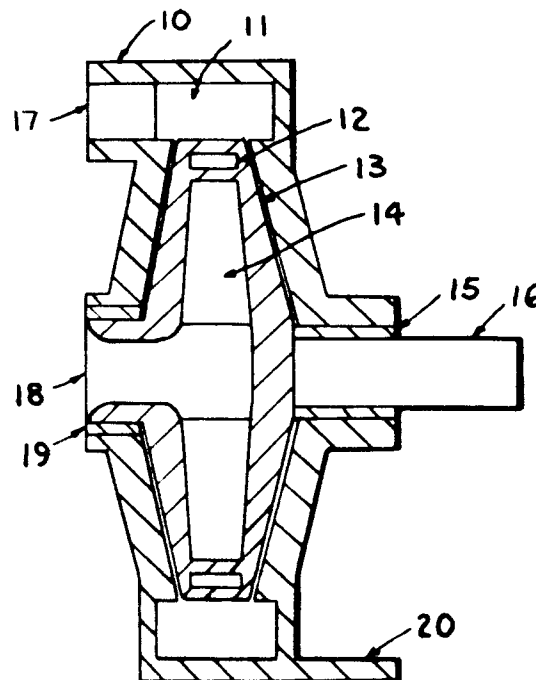
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7 Claims, 3 Drawing Figures



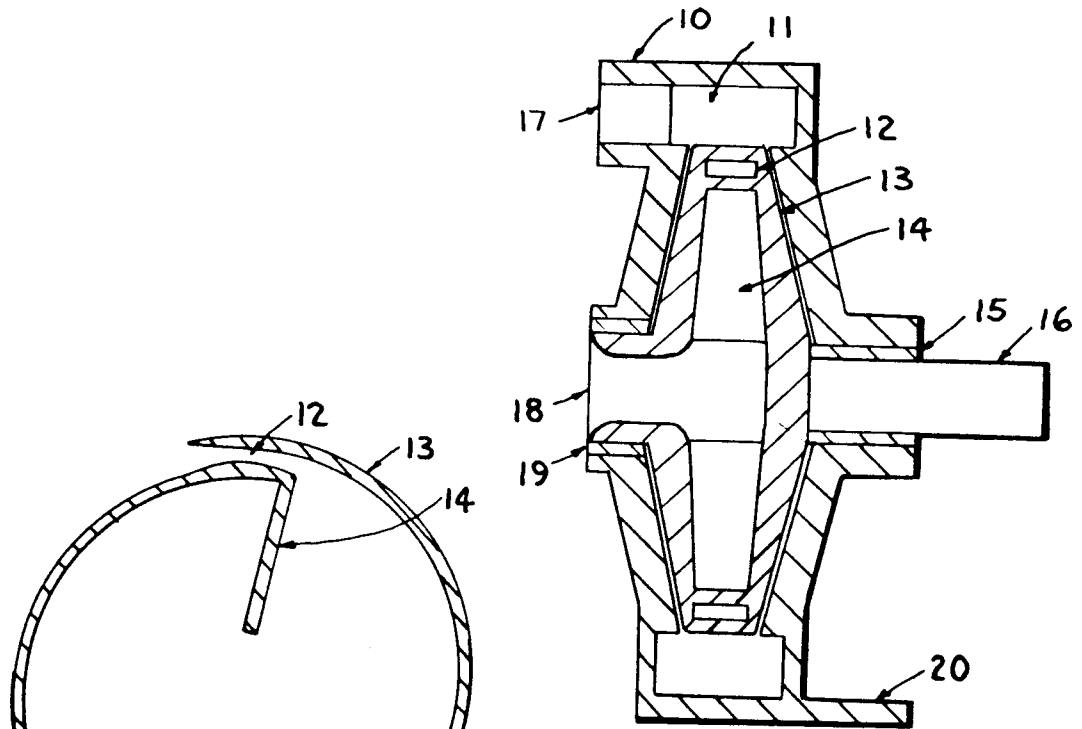


FIG. 1

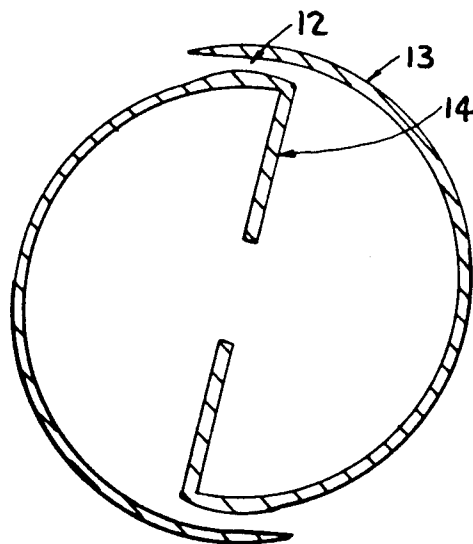


FIG. 3

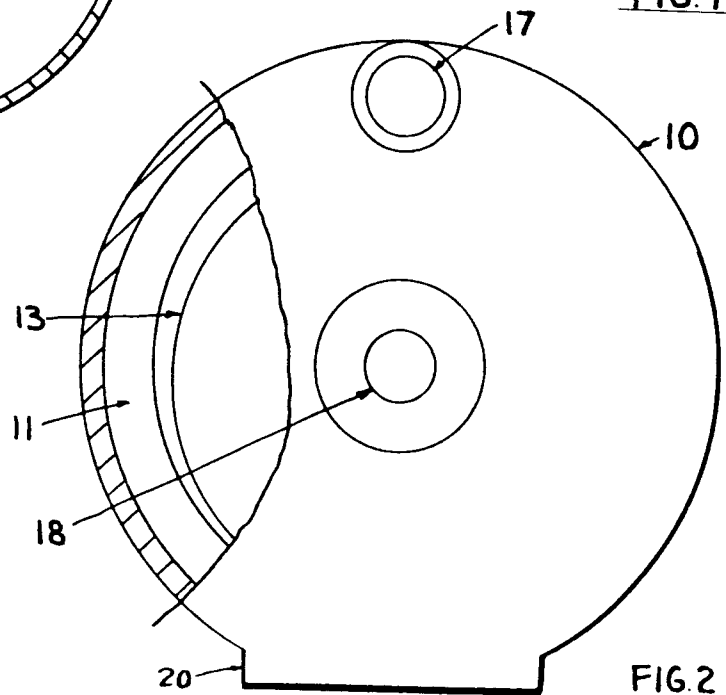


FIG. 2

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REACTION TURBINE

This invention relates generally to power generation devices, and more particularly to turbines employing a fluid reaction on the turbine rotor wheel as the power generating means.

The art of generating power has seen a variety of devices. Reaction turbines have been built with the pressurized fluid being supplied to the rotor inlet and then discharged to a lower pressure via reaction nozzles mounted on a suitable rotating member and discharging said fluid backwards; with said fluid being passed to said nozzles via radial passages so that said fluid is accelerated in said passages with accompanying pressure drop in addition to velocity gain due to centrifugal force.

The main disadvantage of these conventional devices is that they are inefficient due to relatively low fluid exit velocities from said exit nozzles.

FIG. 1 is a cross section of the turbine, and FIG. 2 is an end view of the same unit with a section removed to show interior.

FIG. 3 is a cross section of the rotor wheel showing vanes and nozzles.

It is an object of this invention to provide an improved reaction turbine power generation means wherein the pressurized entry fluid to said turbine is further pressurized within said turbine wheel before being discharged from said wheel via reaction nozzles mounted on said wheel near its periphery and discharging backward, in a direction away from the direction of rotation.

Referring to FIG. 1, therein is illustrated a reaction turbine. 10 is casing, 11 is space around rotor for collecting discharged fluid, 12 is rotor discharge fluid passage, 13 is rotor, 14 is rotor internal vane, 15 is shaft bearing and seal, 16 is rotor shaft, 17 is fluid discharge opening, 18 is fluid inlet to rotor wheel, 19 is bearing and seal and 20 is unit base.

In FIG. 2, an end view of the turbine is shown. 10 is casing, 17 is fluid discharge opening, 13 is rotor, 11 is fluid space, 18 is fluid inlet, 20 is unit base.

In FIG. 3, a cross section of the rotor is shown. 13 is the rotor, 12 is fluid exit nozzle, 14 is rotor internal vane.

In operation, fluid enters said turbine via inlet 18, and is passed to the interior of the rotating rotor. The fluid is pressurized within said rotating rotor by centrifugal action on said fluid by said rotor, with vanes 14 assuring that said fluid will rotate with said rotor. The pressurized fluid is then passed via said exit nozzles to a lower pressure of the discharge; said fluid when passing through said exit nozzles will generate torque on said rotor wheel; said torque is then passed to said rotor shaft and from there as the useful work output of said turbine.

Said fluid exit nozzles are sized and shaped to provide for highest attainable fluid velocity from said nozzles. Said nozzles may be either converging or converging-diverging type as required for the fluid being used.

The unit casing may be fitted closely to the side walls of the rotor wheel as indicated in FIG. 1; this will allow the rotor wheel to partially evacuate the space between the rotating rotor and the casing thereby reducing losses due to fluid friction.

Fluids being used to drive this reaction turbine may be either liquid, gaseous or gases containing liquids in

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various proportions. Also, by suitably shaping said exit nozzles, a fluid that is liquid within said rotor and changes to liquid-gas mixture while passing said nozzles, may be used. Efficiencies that are particularly good may be obtained with gaseous fluids that have a high specific heat value at constant pressure, such as hydrogen, methane or ammonia.

The rotor wheel may have one or more exit nozzles, as desired. Further, more than one unit may be employed with a single fluid stream with the fluid being reduced in pressure in one unit, and then passed to the next unit where the pressure is reduced further; this may be necessary to limit rotor speeds when using high pressure fluids.

In FIG. 1, one fluid entry 18, is shown. The rotor wheel may be provided with entry ports on both sides if desired.

What is claimed is:

1. A device for generating power responsive to flow of fluid therethrough comprising:

a. a power shaft journaled for rotation in a suitable support for transmission of power;

b. a rotating compressing centrifuge rotor carried on said shaft for generating said power; said compressing centrifuge rotor including:

i. an inlet port disposed adjacent the center of said compressing centrifuge rotor for entrance of a fluid thereinto;

ii. a plurality of interior vanes defining passageways therebetween; said passageways diverging as they traverse radially outwardly of said rotor and communicating with said inlet port and having at any radius a first, large cross sectional area and having disposed at the peripheral portion thereof respective obstructions to the flow of fluid to ensure that said fluid flowed into said compressing centrifuge rotor is moved outwardly at a low radial velocity component and attains the same rotational speed as said compressing centrifuge rotor so as to be subjected to a centrifugal force field sufficient to elevate its pressure to form a high pressure fluid at the peripheral portion of said centrifuge rotor;

iii. a plurality of reaction nozzles disposed at the radially outermost portion of said passageways with at least one reaction nozzle communicating with each said passageway; each said reaction nozzle having a second cross sectional area less than said first cross sectional area of its said passageway to ensure that said fluid is compressed by compressing centrifuge action at the operational high rotational speeds of said compressing centrifuge rotor; said reaction nozzles being disposed tangentially of said compressing centrifuge rotor and in a direction opposite the direction of rotation of said rotor so as to direct the effluent fluid tangentially of said compressing centrifuge rotor for developing torque and power that is proportional to a reaction force to the force effecting the velocity difference between the tangential velocity of the effluent fluid and the tangential velocity of said reaction nozzles and, consequently, said compressing centrifuge rotor; said rotor having heavy duty construction sufficient to withstand high rotational speeds;

c. a casing sealingly surrounding said compressing centrifuge rotor; said casing defining a passageway

- surrounding said compressing centrifuge rotor for collection of said effluent fluid; and having respective inlet and outlet passageways; and
- d. bearing means and seal means disposed intermediate and sealingly engaging said shaft and said casing. 5
2. The device of claim 1 wherein said casing closely and conformingly receives the ends of said centrifuge rotor such that the high rotational speed of said compressing centrifuge rotor partially evacuates the space 10 between said ends and said casing and effects a reduced friction at operational high rotational speeds.
3. The device of claim 1 wherein said reaction nozzles are at least converging and are sized and shaped to provide for the highest attainable exit velocity of said 15 effluent fluid from said reaction nozzles.
4. The device of claim 3 wherein said reaction nozzles are converging-diverging for effecting isentropic expansion of said effluent fluid.
5. The device of claim 1 wherein said fluid is a gas 20 having a high specific heat value at constant pressure, and is being flowed through said device.
6. A method of generating power in response to flow

- of a fluid through a device wherein a power delivery shaft is connected with a centrifuge rotor comprising the steps of:
- a. flowing said fluid into said centrifuge rotor and while within the confines of said rotor subjecting said fluid to a centrifugal force field via compressing centrifuge action with low radial velocity component to effect a high pressure fluid at the peripheral portion of said rotor; and
- b. passing said high pressure fluid out of reaction nozzles that are oriented tangentially and rearwardly from the direction of rotation of the periphery of said compressing centrifuge rotor to provide a tangential reaction force to the periphery of said rotor such that a reaction force to said fluid being passed out of said reaction nozzles is imparted to said rotor to deliver power to said shaft.
7. The method of claim 6 wherein said high pressure fluid is expanded isentropically as it is flowed out of said reaction nozzles for a greater velocity and greater reaction force.

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[54] **ROTARY COMPRESSOR WITH COOLING**

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Attorney—Wofford, Felsman & Tails

[22] Filed: **Jan. 11, 1972**

[21] Appl. No.: **216,929**

[52] U.S. Cl. **415/116, 415/178, 415/180,**
126/247

[51] Int. Cl. **F04d 31/00**

[58] Field of Search 415/81, 80, 116,
415/147, 178, 180

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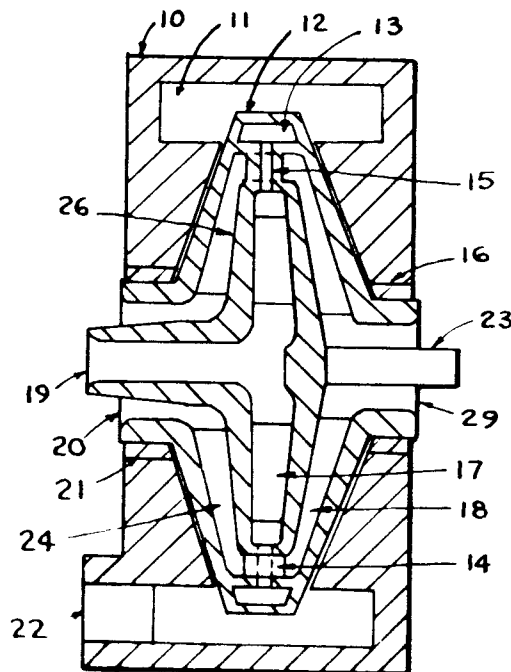
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[57] **ABSTRACT**

A method and apparatus for compressing fluids by employing a rotating rotor wherein said fluid compressed to a pressure that is higher than the unit exit pressure; said fluid is discharged from said rotor via exit nozzles discharging backward; said fluid being cooled within said rotor by employing a second fluid that may be either compressible or non-compressible type. Specific fluids to be used that are disclosed are air as the fluid to be compressed, and water as the second fluid. The device may also be used to produce refrigeration and heating, either with accompanying pressurization of said fluid, or without pressurizing of said fluid.

4 Claims, 3 Drawing Figures



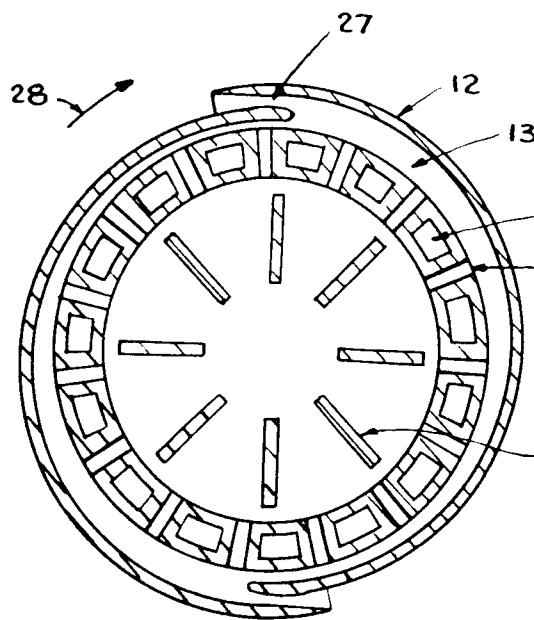


FIG 3

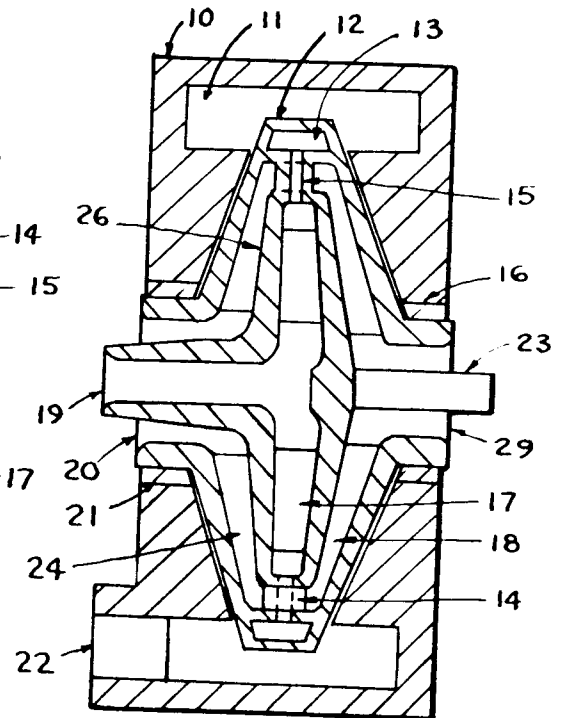


FIG 1

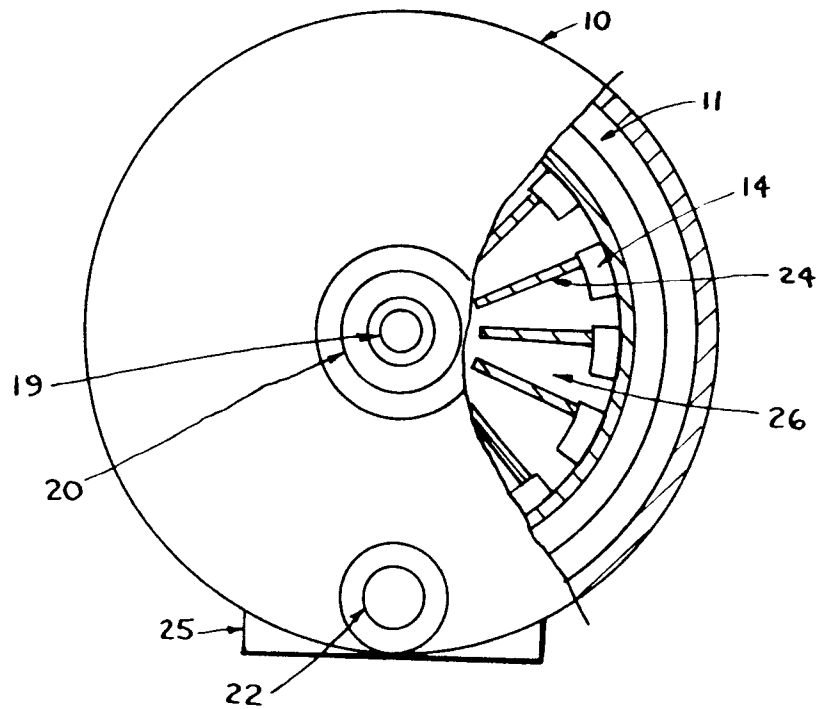


FIG 2

ROTARY COMPRESSOR WITH COOLING

BACKGROUND OF THE INVENTION

This invention relates generally to devices for compressing gases by employing centrifugal force to compress said gas within a rotating rotor.

The art of compressing gases has seen many devices. In some of those devices, a gas is accelerated within a rotating rotor passage and then discharged from said passage usually radially outward, and then said gas is compressed in a diffuser where the kinetic energy of said fluid is converted to pressure.

The main disadvantage of these conventional compressors is that their efficiency is rather poor due to friction to the high speed passage of said gas within said rotor, and losses in converting said kinetic energy to pressure in said diffuser.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of the device, and

FIG. 2 is an end view of the same unit shown in FIG. 1 with a portion removed to show interior details.

FIG. 3 is a section of the rotor being used in this device.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and apparatus for compressing gases wherein said gas is compressed within a rotating rotor to a pressure that is normally higher than the discharge pressure from the unit, with cooling applied to said gas during compression by circulating another fluid in heat exchange relationship with the fluid to be compressed, and discharging said gaseous fluid from said rotor via nozzles oriented to discharge backward.

Referring to FIG. 1, therein is shown a cross section of the compressing device. 10 is casing, 12 is rotor, 11 is a space around rotor, 13 is rotor outer fluid passage, 15 is rotor internal fluid passage for fluid to be compressed, 16 is rotor bearing and seal, 23 is shaft, 17 are rotor internal vanes assuring that the fluid will rotate with said rotor and also serving as heat exchange members, 18 are rotor internal vanes, 14 is a fluid passage for the cooling fluid, 22 is compressed fluid discharge, 24 are rotor internal vanes, 21 is rotor bearing and seal, 20 is cooling fluid inlet, 19 is fluid inlet for the fluid to be compressed, 26 is rotor dividing wall, and 29 is cooling fluid outlet.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated. 10 is casing, 11 is fluid space, 14 is cooling fluid passage, 24 is internal vane, 26 is rotor wall, 19 is fluid inlet, 20 is cooling fluid inlet, 25 is unit base, and 22 is compressed fluid outlet.

In FIG. 3, a section of the rotor for this device is shown. 12 is said rotor, 13 is fluid passage for the fluid being compressed, 14 is cooling fluid passage, 15 is passage for the fluid being compressed, 17 is a vane, 27 is exit nozzle for the fluid being compressed, 28 indicates rotation direction of said rotor.

In operation, the fluid to be compressed enters the said rotor via opening 19, passes to the rotor and is accelerated to the rotor speed with vanes 17 assuring that said fluid will rotate with said rotor. Vanes 17 will also serve as heat exchange members. Said fluid is then passed via passages 15, where the cooling fluid will remove additional heat from the said fluid to be compressed, after which said fluid will be collected in pas-

sage 13, and said fluid will then pass to exit nozzles 27, and be then discharged to collection space 11 within said casing, and will then pass to exit opening 22. Said cooling fluid will enter said rotor via opening 20, be passed through passages 14, and then be discharged via opening 29.

The fluid to be compressed may be a compressible gas such as air, hydrogen, methane or other similar fluid. Also, fluids containing some liquid may be compressed. The cooling fluid may be either a compressible or non-compressible type, such as water, air, or other fluid. Calling the fluid to be compressed as the first fluid, and the cooling fluid as the second fluid, following is the description of the functions of these fluids:

The first fluid is compressed to a higher pressure than the pressure at point 22, by the rotating rotor. Said first fluid normally would gain in temperature when the pressure is increased, but in this device, the second fluid will remove most of such heat so that the compression becomes nearly isothermal. Said first fluid is then discharged from said rotor exit nozzles in a direction that is backward away from the direction of rotation and the fluid jet will produce an impulse on said rotor nozzle, reducing the power input to said rotor from an external source. The velocity of the leaving jet in relation to said nozzle will be determined by the enthalpy difference available between the pressure of the space 13 of the said rotor, and space 11 of said casing. Said nozzles are sized and shaped to provide for highest attainable exit velocity, and said nozzles are either converging or converging-diverging type as required for the fluid.

The second fluid enters said rotor via opening 20 and is then passed to said rotor interior. Said fluid may be either compressible type, such as air, or non-compressible type, such as water. Said second fluid is pressurized within said rotor passage, and for a liquid fluid, there is no appreciable temperature increase due to pressurization, so that the temperature difference between a liquid second fluid and gaseous first fluid will larger, than a temperature difference between a gaseous second fluid and gaseous first fluid. Therefore, for cooling a gaseous first fluid, a liquid second fluid is desirable.

It should be noted that this device also may be used to produce refrigeration, since a gaseous first fluid when compressed isothermally, and then expanded isentropically, will have lower exit temperature at point 22 than in point 19, FIG. 1. Also, this device may be used as a generator of heat, since said second fluid will leave said device at a higher temperature than said second fluid enters. If the device is used to produce refrigeration or heating, the pressure in passage 11 may be lowered to that existing at entry 19, thereby improving the economy of operation by reducing required work input. A combination use for the device is when the unit is used to produce a pressurized gas, while at the same time producing heating and also refrigeration.

To reduce fluid friction on said rotor, the rotor may be closely fitted to said casing, so that by centrifugal action on fluid particles said rotor will partially evacuate space between said rotor walls and said casing, thereby reducing friction.

This device may also be used to heat said first fluid during or after compression, by proper selection of the said second fluid. In this type of application, said second fluid should be a compressible gas, such as air,

while said second fluid may be a fluid such as freon. In this type of application, the temperature gain of the first fluid would be more than the temperature gain for the said second fluid, so that heat transfer will take place from said second fluid to said first fluid with the result that said first fluid will be warmer when being discharged from the unit.

Various well known devices, such as governors, gauges, and the like, are employed with the device of this invention. They do not form a part of this invention and are not further described herein.

What is claimed is:

1. A device for compressing fluids and comprising:
 - a. a rotor for compressing said fluid by centrifugal action on said fluid by said rotor; said rotor having an inlet for said first fluid near the center of said rotor; said first fluid being passed to the interior of said rotor where vanes placed within said rotor interior will assure that said fluid will be rotating with said rotor; said fluid being cooled by a second fluid being circulated in heat exchange relationship with said first fluid within said rotor; said first fluid being discharged from said rotor via exit nozzles oriented to discharge said fluid in backward direction that is away from the direction of rotation; said exit nozzles being sized and shaped to provide for highest attainable exit fluid velocity from said exit nozzles; said exit nozzles being either converging or converging-diverging in shape; said second fluid being either compressible or non-compressible type; said second fluid being passed into said rotor via entry opening near the center of said rotor, being passed from there through passages provided within said rotor to area near the periphery of said rotor and then being passed to area near the center of said rotor and then discharged from said rotor; said rotor being supported by suitable seals and bearings; said rotor being provided with a suitable power shaft;
 - b. a casing to support said rotor and to contain said first fluid; also to provide exit opening for said first fluid;
 - c. a fluid to be compressed and being said first fluid;
 - d. a fluid to be used as the cooling fluid and being said second fluid.
2. The device of claim 1 wherein the casing is closely

fitted to the walls of said rotor to provide for partial evacuation of space between said rotor and said casing by centrifugal action on fluid particles by said rotating rotor thereby reducing friction loss on said rotor.

3. A method of compressing gases and comprising:
 - a. a rotor for compressing said gas wherein a second fluid is circulated to cool said gas during and after compression to produce essentially isothermal compression of said gas;
 - b. discharging said gas via exit nozzles oriented to discharge said gas in backward direction; said nozzles being shaped to obtain highest attainable exit velocity for said gas; employing impulse produced by the said gas when leaving said exit nozzles to reduce the work required to rotate said rotor;
 - c. employing a second fluid as coolant within said rotating rotor; said fluid being supplied to said rotor via openings near center of said rotor, and being discharged via openings near the center of said rotor thereby reducing work required to circulate said second fluid within said rotor.
4. A method of heating a second fluid in a device in which a gas is compressed comprising:
 - a. heating said gas by compression in a rotating continuous flow centrifuge rotor;
 - b. flowing said second fluid at an adjustable flow rate that is less than that required to obtain isothermal compression of said gas in said rotor and that is sufficiently reduced to attain the desired temperature on the effluent second fluid, in heat exchange relationship with said first fluid within said rotating rotor to heat said second fluid to said desired temperature; said second fluid being supplied to said rotor via openings near the center of said rotor, and being discharged via openings near the center of said rotor thereby reducing work required to circulate said second fluid within said rotor; and
 - c. discharging said gas via exit nozzles oriented to discharge said gas substantially tangentially rearward in a direction opposite the direction of rotation of said rotor such that the reaction force induces a torque to rotate said rotor and reduces the work required therefore; said nozzles being shaped to obtain the highest attainable exit velocity for said gas.

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[54] REACTION ROTOR TURBINE

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[22] Filed: Sept. 30, 1971

[21] Appl. No.: 185,060

[52] U.S. Cl. 415/1, 415/80

[51] Int. Cl. F01d 1/00, F01d 1/06

[58] Field of Search 415/80-87, 1

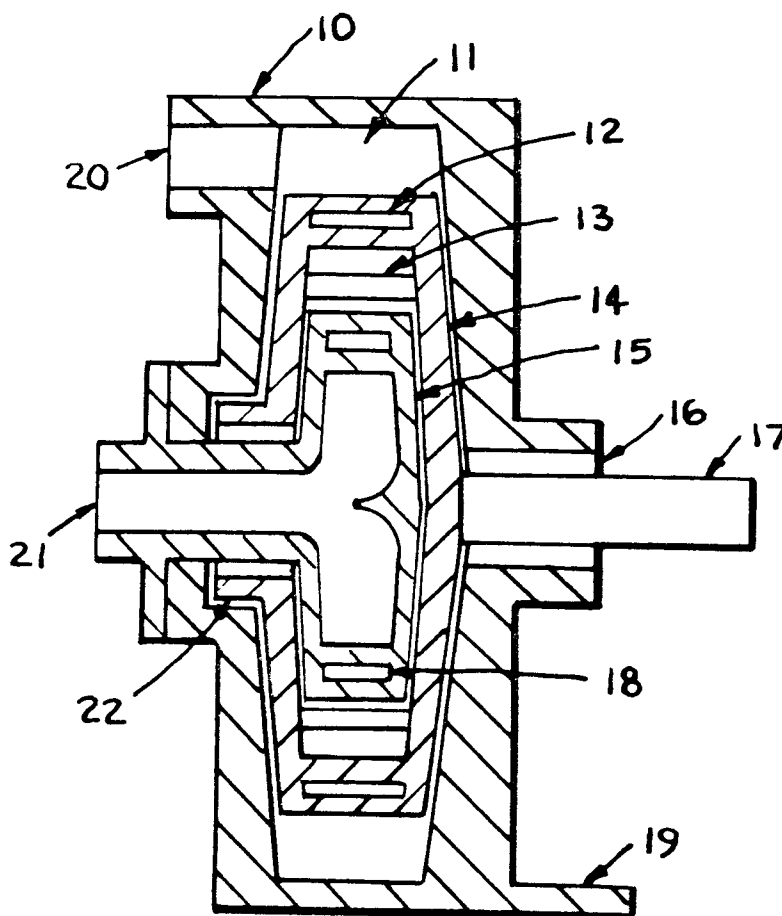
[57] ABSTRACT

A method and apparatus for a reaction type turbine rotor, employed in power generation devices, such as steam turbines, hydraulic turbines, or gas turbines. The pressurized fluid is supplied to within said turbine reaction rotor by employing a fluid nozzle arranged to discharge within said wheel in a direction that is approximately the same as the direction of rotation of said rotor wheel. The fluid is then discharged from the rotating rotor wheel nozzles backward away from the direction of rotation; said fluid then will generate a torque on said rotor wheel; said torque is then passed to the rotor shaft as the work output of the device.

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6 Claims, 4 Drawing Figures



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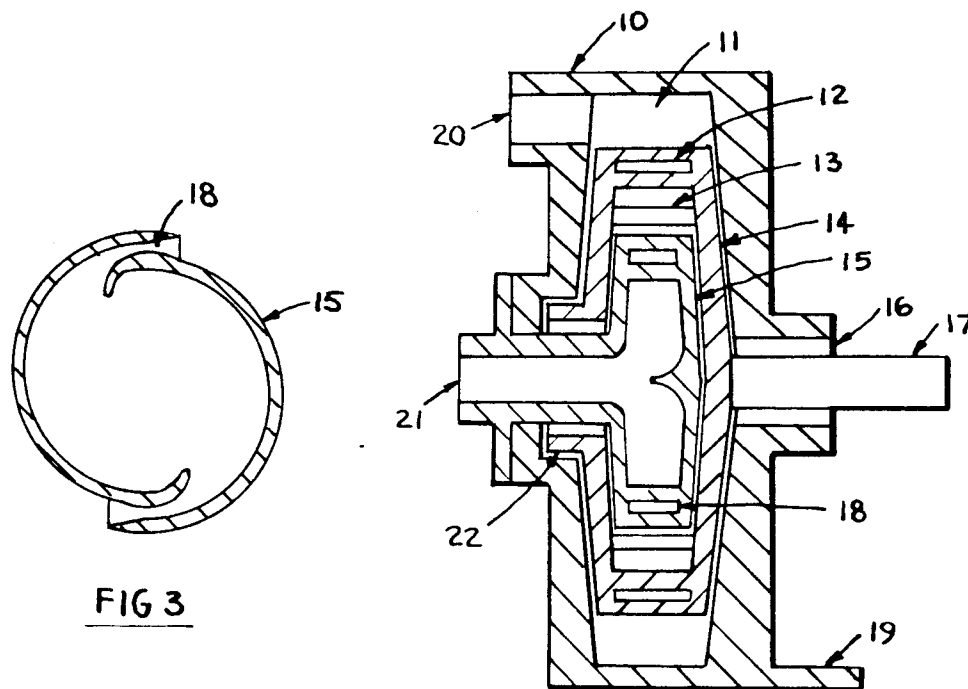


FIG. 1

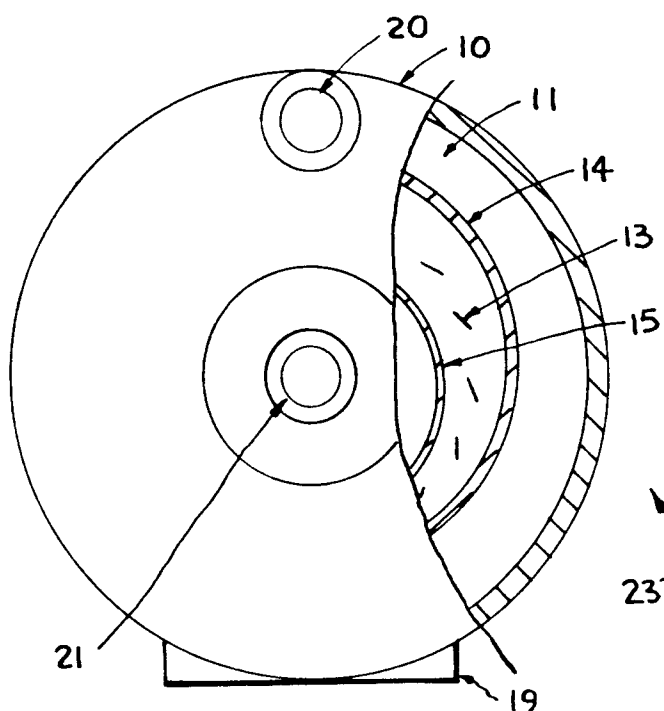


FIG. 2

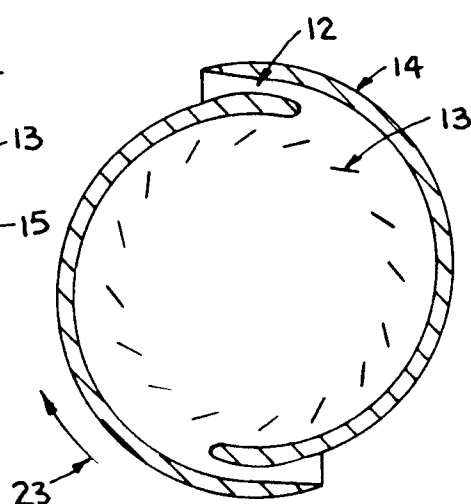


FIG. 4

INVENTOR.

BY Michael Eskel

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REACTION ROTOR TURBINE

BACKGROUND OF THE INVENTION

This invention relates to power generation devices, and more particularly to turbines employing fluid reaction on the turbine rotor wheel as power generating means.

The art of generating power has seen a variety of devices. Reaction turbines have been built with the pressurized fluid being supplied to the rotor inlet and then discharged to a lower pressure via reaction nozzles mounted on the rotor periphery and discharging said fluid backwards.

The main disadvantage of these conventional devices is that the rotating rotor wheel is required to accelerate said fluid first to the tangential speed of said rotor, before discharge, which results in a reduced power output for the device, and decreases the efficiency of the device.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of a power generation device employing the reaction rotor, and the tangential fluid feeder.

FIG. 2 is an end view of the device shown in FIG. 1.

FIG. 3 is a sectional end view of the tangential fluid feeder.

FIG. 4 is a sectional end view of the reaction rotor.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide an improved means for generating power by employing a reaction rotor in a turbine, with a tangential fluid feeder to said rotor; these two items forming a stage in a power generation turbine; said stage may be employed either in a steam turbine, a gas turbine or in a hydraulic turbine installation.

Referring to FIG. 1, therein is illustrated a cross section of a power generation device employing said reaction rotor, and tangential fluid feeder. 10 is a casing, 11 is a fluid collection space around the reaction rotor, 12 is a discharge nozzle opening on reaction rotor 14, 13 are fluid guide vanes within said rotor, 16 is a bearing and seal for reaction rotor shaft 17, 18 is a fluid feeder nozzle opening, 19 is base for casing 10, 20 is fluid outlet, 21 is inlet to stationary fluid feeder and 22 is a bearing and seal.

In FIG. 2, 20 is fluid outlet, 21 is fluid inlet, 10 is casing, 11 is fluid space, 14 is rotor, 13 are fluid guide vanes within rotor 14, 15 is the tangential fluid feeder.

In FIG. 3, an end sectional view is shown of the tangential fluid feeder. 18 is a fluid exit nozzle, 15 is the feeder body.

FIG. 4 illustrates the reaction rotor. 14 is the rotor body, 12 are fluid exit nozzle, 13 are fluid guide vanes, and 23 is an arrow indicating the direction of rotation for the rotor.

In operation, the fluid enters the tangential fluid feeder at inlet 21, and is passed to the rotor 14 via exit nozzles 18. Said fluid is accelerated in said exit nozzles, with an accompanying pressure reduction. Within said reaction rotor, said fluid is compressed. Said fluid is then discharged through reaction nozzles mounted near the periphery of said rotor, with said nozzles being arranged to discharge said fluid backward in a direction away from the direction of rotation of said rotor wheel; the velocity of said fluid from said nozzles being depen-

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dent on the pressure differential between the exit end and the entry end of said nozzles, and the fluid being used. Torque is produced on said rotor wheel by reaction to the force of said fluid when said fluid passes through said nozzles; said torque is then transmitted to the rotor shaft and from there as the useful work output of the reaction rotor turbine stage. It should be noted that the fluid pressure for this device is normally higher upstream of the feeder inlet, than downstream of the reaction rotor nozzle outlet.

The feeder nozzles, 18 in FIG. 3, are normally converging, or converging-diverging, in shape, depending of the fluid being used. The purpose of these nozzles is to provide for highest attainable exit velocity from these nozzles for the pressure and enthalpy differentials available between entry and exit conditions for these nozzles. The feeder unit, FIG. 3, is shown to be round in shape; this shape was selected to provide for least fluid resistance on the exterior of said unit. Other feeder shapes may be used, such as pipes, with the nozzles attached at the tips. The number of nozzles for the unit shown in FIG. 3, may be as desired.

The rotor unit shown in FIG. 4, may have one or more nozzles, as desired. The nozzles may be either converging, or converging-diverging shape, as required for the fluid being used. The fluid velocity leaving these nozzles, related to the nozzle velocity, should be highest attainable, for the pressure and enthalpy differential available between the entry and exit ends of said nozzles. The guide vanes, 13, may be oriented as desired; these vanes may not be required with all fluids; purpose of these vanes is to minimize turbulence within said rotor cavity.

The turbine stage of this invention may be used with fluids such as steam, air, various other gases and with liquids such as water. In hydraulic power plants, using water from a dam as the pressurized entry fluid, the unit may be conveniently placed with the rotor horizontal and the axis vertical, with the water entering inlet 21 from above. In most installations, it would be desirable to control the unit by having suitable adjustable inserts at the nozzles 18 to control fluid flow; similarly, the nozzles of rotor 14 may be adjustable to control fluid flow; devices to provide for these controls and adjustments are not part of this invention and are not further described herein.

To explain further the operation of the reaction rotor stage of this invention, it should be noted that by accelerating said fluid in the said stationary feeder nozzles 18 to the tangential speed of the said rotor 14, there is no work required to accelerate said fluid to said speed in said rotor; except to the extent that the tangential velocity of said rotor differs from the velocity of said entering fluid. Therefore, the work output produced when said fluid issues from said rotor 14 nozzles 12, is nearly all available at the rotor shaft to be passed to an external load as useful work. This is in contrast to an ordinary reaction rotor, where said fluid has to be first accelerated to the rotor speed, before said fluid is discharged from the reaction nozzles with the resultant reduction in useful work output.

The velocity of the fluid exiting from the fluid feeder nozzle opening 18 is higher than the velocity of the rotor at that radial distance. Preferably, the velocity of the fluid is at least as great as the velocity of the rotor at that radial distance. The greater the velocity of the fluid flowing out of the fluid feeder nozzle openings 18,

the greater work that is realizable from the reaction rotor turbine employed as a power generation device.

Appropriate and well known equipment, controls and governors, are employed with the device described hereinbefore. They do not form any part of this invention and are not further described herein.

What is claimed is:

1. A turbine stage for power generation comprising:
 - a. a reaction rotor for producing said power; said rotor being rotated at a predetermined speed; said rotor having reaction nozzles mounted near the periphery of said rotor; said reaction nozzles being oriented to discharge a fluid backward in a direction that is away from the direction of rotation of said rotor; said rotor being hollow and having a first cross sectional area at any radial distance that is greater than a second cross sectional dimension of said reaction nozzles so as to effect a peripheral pressure that is greater than the pressure at the inlet to the rotor downstream of a set of fluid feeder nozzles; said reaction nozzles being at least converging and sized and shaped to provide for maximum exit velocity attainable for said fluid from said nozzles relative to said rotor depending on the pressure and enthalpy differential available between entry and exit ends of said nozzles;
 - b. a fluid feeder; said feeder being stationary and mounted concentrically interiorly of said rotor; said feeder being connected with a source of said fluid at an elevated pressure for supplying fluid to said reaction rotor; said feeder being equipped with fluid feeder nozzles arranged to discharge said fluid in a direction that is substantially the same as the tangential and at a speed greater than the speed of the reaction rotor in the area where said fluid enters said rotor; said fluid feeder nozzles being at least converging and sized and shaped to provide for highest attainable velocity for said fluid depending on the pressure differential available between entry and exit ends of said fluid feeder nozzles;
 - c. a casing for providing support for said reaction rotor and said fluid feeder;
 - d. a shaft connected with said rotor and journaled in said casing for supporting said rotating reaction rotor;

- f. bearing means disposed about said shaft for reducing friction; and
- g. fluid being flowed through said turbine stage from an inlet pressure upstream of said fluid feeder to a discharge pressure downstream of said reaction nozzles; said inlet pressure being greater than said discharge pressure; said fluid being flowed from said fluid feeder nozzles at a speed greater than the tangential velocity of said rotor at the same radial distance for effecting a greater power output on said shaft.
2. The turbine stage of claim 1 wherein said fluid is water.
3. The turbine stage of claim 1 wherein said fluid is a hot fluid such as steam for converting heat to power.
4. The turbine stage of claim 1 wherein said fluid is a liquid.
5. The turbine stage of claim 1 wherein said rotor has vanes disposed at a predetermined radial distance intermediate said fluid feeder and said reaction nozzles for minimizing turbulence in said rotor and improving the efficiency of compression of said fluid in said rotor.
6. A method of generating power in a reaction turbine comprising the steps of:
 - a. accelerating a fluid to a first velocity and direction with respect to a radial extending outwardly from a central longitudinal axis of a fluid feeder;
 - b. passing the accelerated fluid at said first velocity into a rotating rotor; said first velocity being greater than the tangential velocity of the rotor at the same radial distance from said axis; said direction being substantially tangential to said radial velocity direction;
 - c. causing said fluid to flow in an arcuate radially outward path with sufficient angular velocity to cause sufficient centrifugal force to pressurize said fluid within said rotor such that the pressure of said fluid at the periphery of said rotor is greater than the pressure at the inlet to said rotor; and discharging said fluid through reaction nozzles mounted at the periphery of the rotating rotor in a direction that is opposite to the tangential direction of rotation for producing thrust and generating torque on the rotor and its shaft for producing power efficiently.

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[54] COMPRESSING CENTRIFUGE

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[21] Appl. No.: 140,124

[52] U.S. Cl. 415/1, 415/80, 415/83,

415/147

[51] Int. Cl. F01d 1/06, F01d 1/22

[58] Field of Search 415/147, 80, 83

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Primary Examiner—Martin P. Schwadron

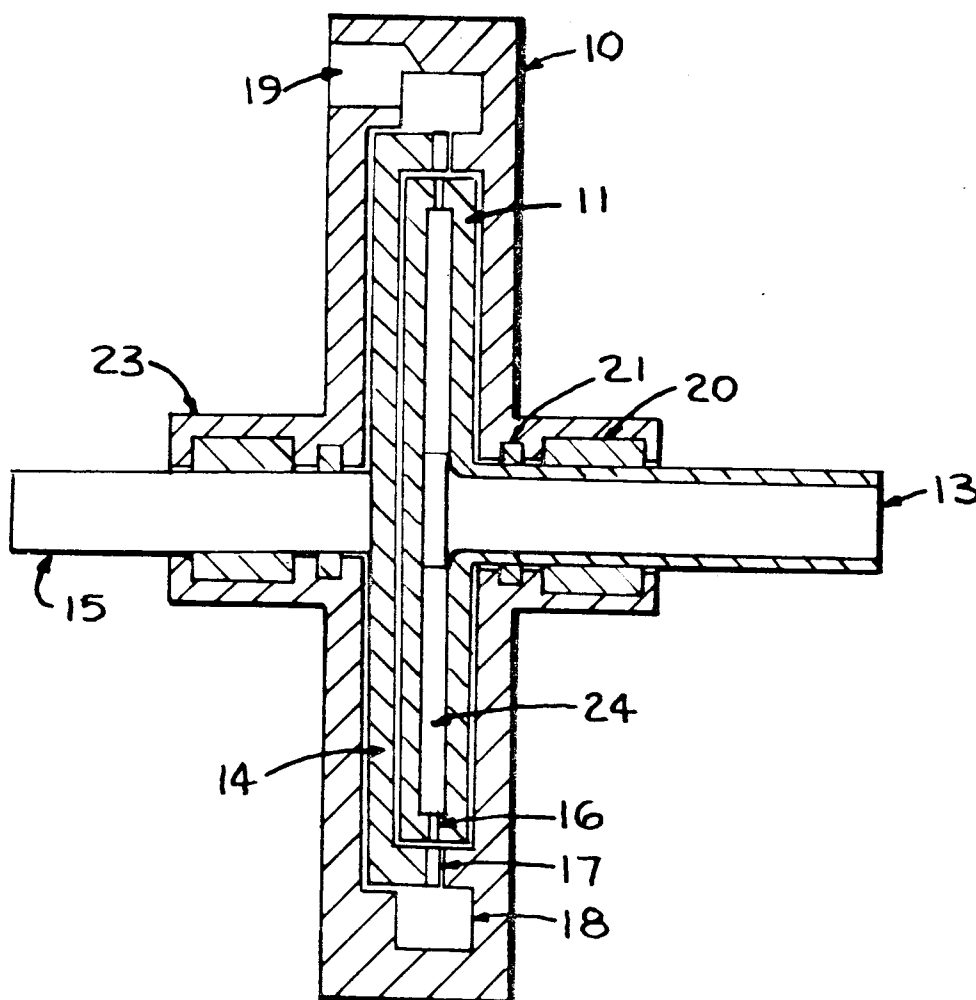
Assistant Examiner—Allen M. Ostrager

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[57] ABSTRACT

Method and apparatus for compressing dry or wet gases wherein the gas is compressed in a high speed rotor and discharged from said rotor in essentially compressed state to a secondary rotor where the kinetic energy of the fluid stream is converted to work; this work can be then used to decrease the work input to the primary rotor, resulting in an improved efficiency for the machine. Dry gases, such as air, may be compressed; also, wet gases or vapors, with a predetermined amount of liquid, such as gas-liquid mixture of propane, may be compressed and liquefied within the said primary rotor with the fluid being a liquid when reaching the secondary rotor and being discharged as a liquid from the machine. Further, liquid may be added to a dry gas in a predetermined amount, for example, water added to air, to absorb some heat during compression and thereby decrease primary rotor speed.

13 Claims, 4 Drawing Figures



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PATENTED SEP 25 1973

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SHEET 1 OF 3

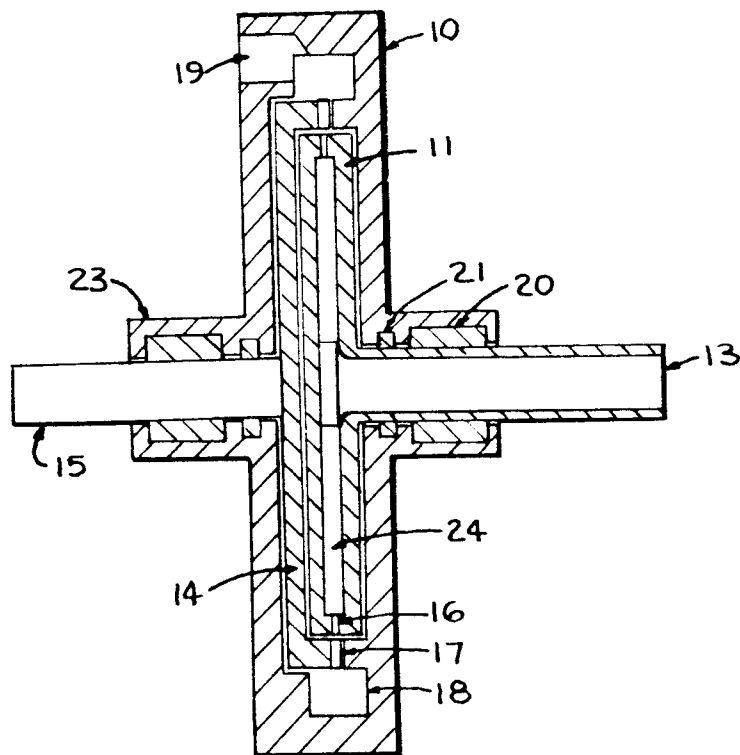


FIG. 1

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BY *Michael Erskeli*

PATENTED SEP 25 1973

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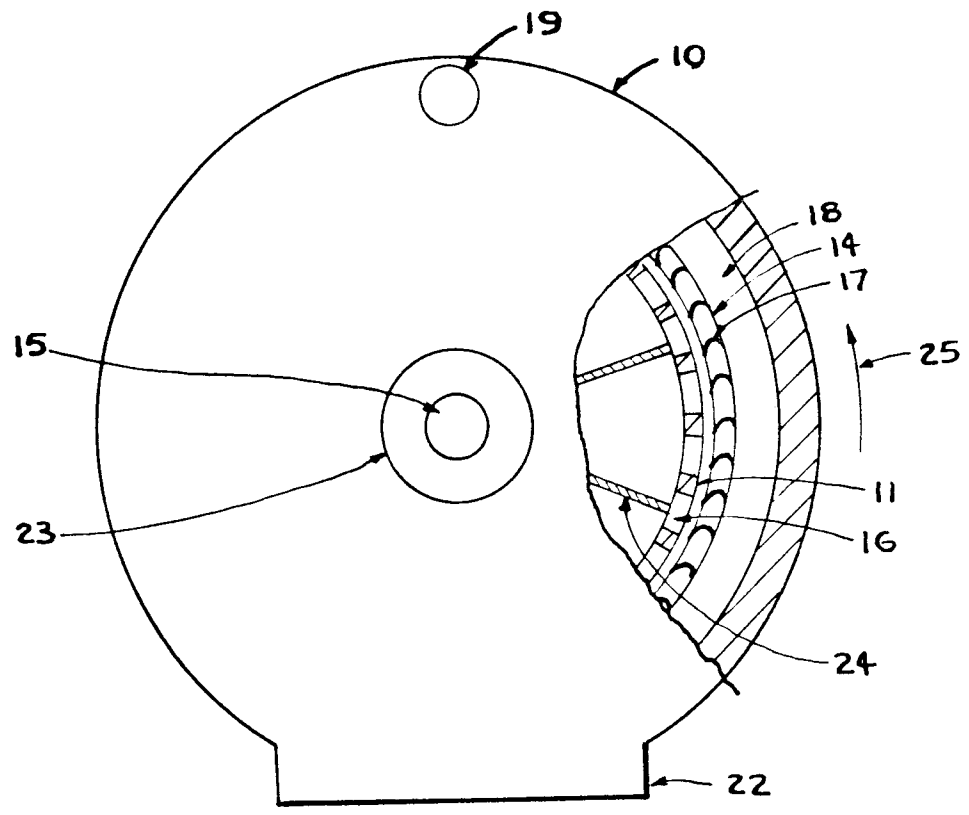


FIG. 2

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PATENTED SEP 25 1973

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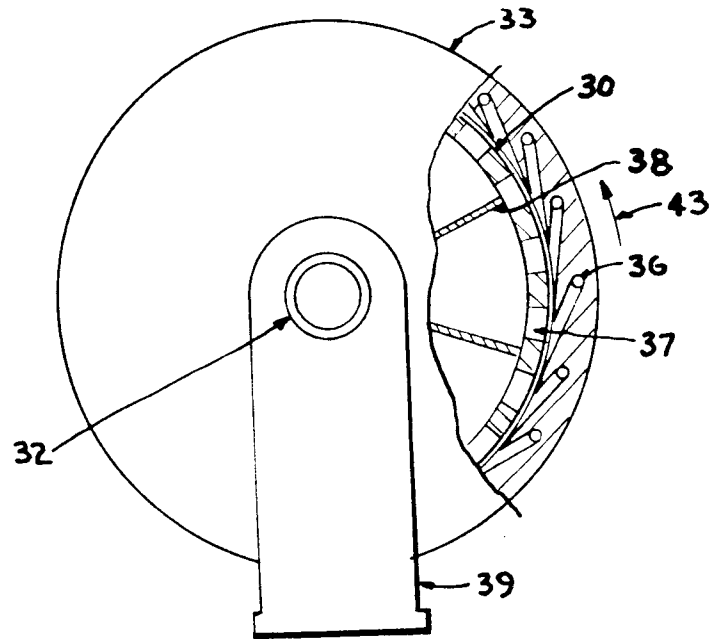


FIG. 3

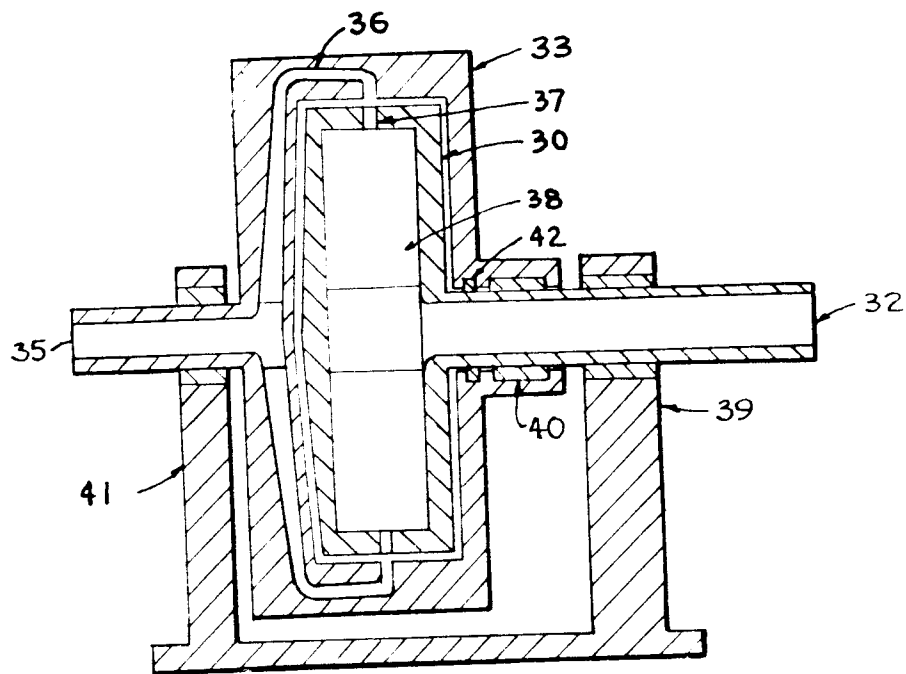


FIG 4

INVENTOR.

BY *Michael Eskeli*

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COMPRESSING CENTRIFUGE

BACKGROUND OF THE INVENTION

The invention relates generally to devices for compressing gases; either as a dry gas or as a wet gas with an amount of liquid fed to the compressor with the gas.

The art of compressing gases has seen variety of devices. One large group of such devices uses centrifugal force to accelerate the gas in an impeller, then throwing the gas to the diffuser where said gas is compressed by converting the kinetic energy imparted to the gas by the impeller, to pressure.

The main disadvantage of these conventional compressors is their poor efficiency.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of one form of the compressor.

FIG. 2 is an end view of the same compressor shown in FIG. 1, as well as an embodiment of invention.

FIG. 3 is an end view of another form of the compressor and FIG. 4 is a sectional view of the same compressor, as well as an embodiment of this invention.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and an apparatus to compress gases, either dry, or gases containing liquids, in which the gas is compressed within a rotor to approximately to its final pressure, after which the gas is passed from the rotor via suitable openings to a section where the kinetic energy contained by the the gas is converted to work with only minor changes in the pressure of the gas. After the kinetic energy is absorbed by the secondary rotor, the gas is passed to the machine gas outlet in compressed condition and at a pressure that is higher than the inlet pressure of said gas.

It is also an object of this invention to provide means for compressing gases with suitable properties, wherein the gas is liquefied within the primary rotor, after which the liquid is passed from the primary rotor to the secondary rotor wherein the kinetic energy of the liquid is converted to power; after the liquid has been slowed to a suitable velocity, it is passed to the outlet from said compressor in liquid form.

Referring to FIG. 1, therein is illustrated a sectional view of one form of said compressor. 10 is a stationary compressor casing, 11 is the primary rotor, equipped with openings 16 on the periphery and with vanes 24 defining cavities within the rotor for assuring that the fluid within said rotor will rotate at same velocity as the rotor. Gas is admitted to said primary rotor via hollow shaft opening 13; said gas may be dry or may have a predetermined amount of liquid with it. After compression of the fluid by centrifugal action within the primary rotor, said fluid leaves the rotor through openings 16, and is passed to openings 17 in the secondary rotor. The rotational velocity of the secondary rotor is maintained to convert the kinetic energy contained in the fluid, to work, in said secondary rotor. In passage 17, vanes are placed to deflect the said fluid stream; the design of these vanes and sizing of the passages is conventional and is not further described herein. 14 is the secondary rotor, 15 is the work output shaft, 18 is an annular space for collecting the fluid in its compressed condition, 19 is fluid outlet, 20 is a shaft bearing, 21 is a shaft seal and 23 is a bearing housing.

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In FIG. 2, an end view of the same compressor is shown. 10 is the stationary housing, 11 is the primary rotor, 14 is the secondary rotor, 24 are vanes in the primary rotor, 16 are openings at the periphery of the primary rotor, 17 are deflection vanes, or buckets, in the secondary rotor, 18 is the annular space for collecting the fluid, 19 is the fluid outlet, 15 is the work output shaft, 23 is the bearing housing, 22 is stationary housing support, and 25 is an indicator arrow to show the direction of rotation of said primary and secondary rotors.

In FIG. 3 an end view of a second form of the compressor is shown; the main difference being the method of converting the kinetic energy contained by the fluid, to work. 33 is the secondary rotor housing, 30 is the primary rotor, 38 are primary rotor vanes, 43 indicates the direction of rotation for the primary and secondary rotors, 36 is a fluid passage in the secondary rotor, 37 is a fluid passage in the primary rotor, 32 is a hollow shaft for fluid input and for work input into said primary rotor, and 39 is a support for the compressor.

In FIG. 4, a sectional view of the compressor shown in FIG. 3, is illustrated. 33 is the secondary rotor housing. The fluid to be compressed enters the compressor via hollow shaft 32 and passes to primary rotor 30, vanes 38 within said rotor are used to accelerate the fluid to the velocity of said rotor. The fluid leaves the said primary rotor via openings 37, and passes to secondary rotor passages 36, where the kinetic and rotational energy of said fluid is converted to work; the said fluid then is passed out from the compressor via hollow shaft 35. 40 is a bearing, 42 is a shaft seal, and 39 and 41 are shaft bearing supports.

The function of the compressor illustrated in FIG. 1 and FIG. 2, is as follows: Gas or gas-liquid mixture, is passed to the primary rotor 11, FIG. 1, where the gas is compressed by centrifugal action within the rotor. Work is supplied to said rotor via shaft 13, causing said rotor and shaft to rotate at high speed. The rotational speed is determined by the density of the gas and by the amount of compression desired. The fluid leaves the rotor via openings 16 at the periphery of said rotor; these openings are designed and sized to retain the fluid within the said primary rotor until the desired amount of compression is reached. After leaving the primary rotor, the fluid will have essentially the same velocity as the primary rotor periphery has; this high velocity fluid is then deflected in the vanes 17 of the secondary rotor. The speed of the secondary rotor is chosen to convert most of the kinetic energy contained in the fluid stream to work, without any major changes in the pressure of the fluid. In the illustrations, one row of vanes or buckets are shown; more than one row may be used, with stationary bucket rows between the moving rows, if desired. The shaft 13 of the primary rotor is connected to a power source, such as an electric motor, and the shaft of the secondary rotor is connected to a load, or may be connected to the primary rotor shaft via a suitable power transfer device, such as a gearbox.

The function of the compressor shown in FIG. 3 and FIG. 4 is similar to that described above; the main difference being in the method of converting the velocity of the fluid leaving the primary rotor, to work. After leaving the primary rotor, the fluid enters into passages 36 in the secondary rotor, where fluid stream is deflected in a curving passage, and passed to the center and out through a hollow shaft. The conversion of kinetic energy to power in the said passage 36 of second-

ary rotor 33, functionally is similar to passages in inward-flow, turbines, or commonly known as Francis turbines. In this type arrangement, there is some pressure loss for the fluid in said passage 36 due to centrifugal forces; however, this pressure loss is not significant due to lower angular speed of the secondary rotor; also, by suitable design of these passages said pressure losses can be minimized.

In the compressing centrifuge described hereinbefore pressures are lowest near the center of the primary rotor and build up so that the pressure is higher near the discharge passageways. Ordinarily, the pressure near the periphery of the primary rotor is higher than the pressure in the casing, or discharge of the compressing centrifuge, to accommodate a pressure drop of the fluid passing through the nozzles, or discharge passageways, of the primary rotor. The fluids passing through the discharge passageways may attain high velocities. There may be employed any of the conventional methods of converting to velocity the pressure drop of the compressed fluid flowing out of the discharge passageways. For example, the primary rotor discharge passageways may be sized and shaped to provide for isentropic expansion of the fluid passing there-through so that the higher than ordinary velocities can be attained. The proper design for such discharge passageways, or nozzles, to effect isentropic expansion is well known and is discussed in standard texts and references such as KENT'S MECHANICAL ENGINEERS HANDBOOK, "Power Volume," J. K. Salisbury, editor, 12th edition, Wylie Engineering, Inc., New York, 1950, chapters 4-03 and 10-02. As discussed therein, the discharge passageways may be slightly converging although the degree of convergence is small for effecting isentropic expansion; the discharge passageways may be nonconverging; or the discharge passageways may be diverging, as illustrated by passageways 16 and 37 in FIGS. 2 and 3.

The compressors shown can be used to compress either dry or wet gases. The compression of dry gases is essentially as described above; and include gases such as air, methane, and others. The compression of gases such as propane, ethylene or other similar gases, where the fluid may exist in both liquid and vapor form at ambient temperatures, can be advantageously accomplished with the compressor described in this invention. The vapor is passed to the compressor via the hollow shaft, with a suitable amount of liquid; the purpose of the liquid is to help condense the vapor, and at the same time absorb the heat of compression and the heat of vaporization of the vapor that is being liquefied within the primary rotor. The fluid in this type of usage will be all liquid when leaving the primary rotor; normally, the primary rotor would have a layer of liquid within, although this liquid layer is not mandatory. That is, the amount of liquid may be less than the amount necessary to provide a liquid layer. On the other hand, an amount of liquid that is greater than the minimum amount necessary to create the liquid layer may be employed. The fluid temperature for this liquid leaving the compressor will be higher than the fluid entry temperature to the compressor due to the heat of vaporization of the gas being added to it. The compressor shown in FIG. 3 and FIG. 4 can advantageously be used to compress liquid-vapor mixtures.

The two arrangements illustrated in the FIGURES differ primarily in the way the kinetic energy contained

by the fluid when leaving the primary rotor, is converted to work. Other arrangements for said conversion may be used, some of which will be briefly listed here:

a. Mount buckets on the secondary rotor; the buckets similar to those used in impulse type hydraulic turbines, commonly known as Pelton wheels.

b. Convert the velocity energy to pressure in a suitable diffuser and then lower the pressure in a suitable engine or gas expander to obtain work; the diffuser and the gas expander may be built within the same housing as the compressor.

Further, to decrease the speed of the primary rotor for a given amount of compression, liquid may be added to the compressor inlet. Example for this would be the addition of water when compressing air; the water would absorb some of the heat of compression and thereby increase the density of the air within the primary rotor.

I claim:

1. A method of compressing a fluid comprising:

a. subjecting said fluid to a centrifugal force field via a primary rotor having vanes defining cavities to ensure that said fluid attains the same rotational speed as said rotor, in a compressing centrifuge to compress said fluid to a pressure that is higher than the discharge pressure from said compressing centrifuge;

b. passing said fluid in its compressed state through discharge passageways that are smaller in cross sectional area than the area of said respective cavities intermediate said vanes upstream thereof and that are located on the periphery of said primary rotor such that centrifuging action is effected and essentially the same velocity as that of the primary rotor periphery is imparted to said fluid from said primary rotor to a secondary rotor in which a large portion of the kinetic energy contained by said fluid is extracted and converted to useful work; and

c. passing said fluid in its compressed state from said secondary rotor to a compressing centrifuge outlet, the pressure of said fluid at said compressing centrifuge outlet being higher than at said compressing centrifuge inlet.

2. A method of compressing a fluid comprising:

a. subjecting said fluid and a liquid forming a gas-liquid mixture to a centrifugal force field via a primary rotor having vanes defining cavities to ensure that said gas-liquid mixture attains the same rotational speed as said rotor, in a compressing centrifuge to compress said gas-liquid mixture to a pressure that is higher than the discharge pressure from said compressing centrifuge; the liquid being fed to the compressing centrifuge at the primary rotor at a predetermined rate to form said gas-liquid mixture; said gas-liquid mixture including any additional liquid formed by condensation and solution of the fluid in the liquid injected there-into because of the centrifuging action and the centrifugal force field;

b. passing said gas-liquid mixture in its compressed state through discharge passageways that are smaller in cross sectional area than the respective said cavities intermediate said vanes upstream thereof and that are located on the periphery of said primary rotor such that centrifuging action is effected and essentially the same velocity as that of the primary rotor periphery is imparted to said

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fluid from said primary rotor to a secondary rotor in which a large portion of the kinetic energy contained in said gas-liquid mixture is extracted therefrom and converted to useful work; and

- c. passing said gas-liquid mixture to the compressing centrifuge outlet, the outlet pressure being higher than the compressing centrifuge inlet pressure.

3. The method of claim 2 wherein at least a portion of the fluid to be compressed is liquefied within the primary rotor and is passed therefrom through the secondary rotor to the compressing centrifuge outlet in its liquid state, and wherein part of the heat of compression is absorbed by the fluid in its liquid state.

4. The method of claim 2 wherein the liquid being added to the fluid at the compressing centrifuge inlet is different from said fluid, the liquid being added for the purpose of absorbing part of the heat of compression of the gas.

5. The method of claim 2 wherein the liquid being added to said fluid at the compressing centrifuge inlet is liquid phase of said fluid.

6. The method of claim 2 wherein said fluid is subjected to isentropic expansion in passing through discharge passageways from said primary rotor for increased kinetic energy.

7. A compressing centrifuge for compressing fluids comprising:

- a. a primary rotor means for subjecting said fluid to a centrifugal force field; said primary rotor means having a space with internal vanes defining respective cavities within said primary rotor for ensuring that any fluid within said primary rotor means rotates with the same velocity as said primary rotor means; said primary rotor means being equipped with means for introducing the fluid to be compressed at the center of said primary rotor means and having suitable discharge passageways adjacent the periphery thereof for discharging the compressed said fluid; said discharge passageways being smaller in cross sectional dimensions than the minimum cross sectional dimension of the associated cavity; said primary rotor means also having

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a shaft for power input needed to effect rotation thereof; and

- b. a secondary rotor means for absorbing and converting to useful work a large portion of the kinetic energy of said fluid in its compressed state leaving the discharge passageways of said primary rotor means; said secondary rotor means having suitable vane means for absorbing energy from the fluid stream as it leaves the discharge passageways of said primary rotor means.

8. The compressing centrifuge of claim 1 wherein said secondary rotor means also serves as a casing with the compressed fluid being discharged via the center shaft thereof, said center shaft also effecting delivery of the power to accomplish said useful work.

9. The compressing centrifuge of claim 7 wherein a casing is provided for said primary and secondary rotor means; said casing having suitable seals and bearings for support of the respective shaft supporting said primary and secondary rotor means and for containing said fluid; said casing having respective passageways through which said fluid may be taken in and discharged respectively.

10. The compressing centrifuge of claim 7 wherein said discharge passageways of said primary rotor means are shaped for effecting isentropic expansion of said fluid passing therethrough.

11. The compressing centrifuge of claim 10 wherein said discharge passageways are at least nonconverging at their discharge end section.

12. The compressing centrifuge of claim 11 wherein said discharge passageways are diverging at their discharge end section.

13. The compressing centrifuge of claim 7 wherein a diffuser passage is provided downstream of the secondary rotor vanes for increasing the pressure of the fluid where only a part of the kinetic energy contained in the fluid stream is converted to work via the secondary rotor means; said diffuser passage effecting some conversion of the kinetic energy to increase the pressure of the fluid at the discharge of said compressor.

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[54] COMPRESSOR WITH COOLING

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[22] Filed: Aug. 10, 1972

[21] Appl. No.: 279,739

[52] U.S. Cl. 415/1, 415/178, 415/211,
62/86

[51] Int. Cl. F04d 29/58, F01d 5/08

[58] Field of Search..... 415/178, 219 B, 1; 62/499,
62/86, 401

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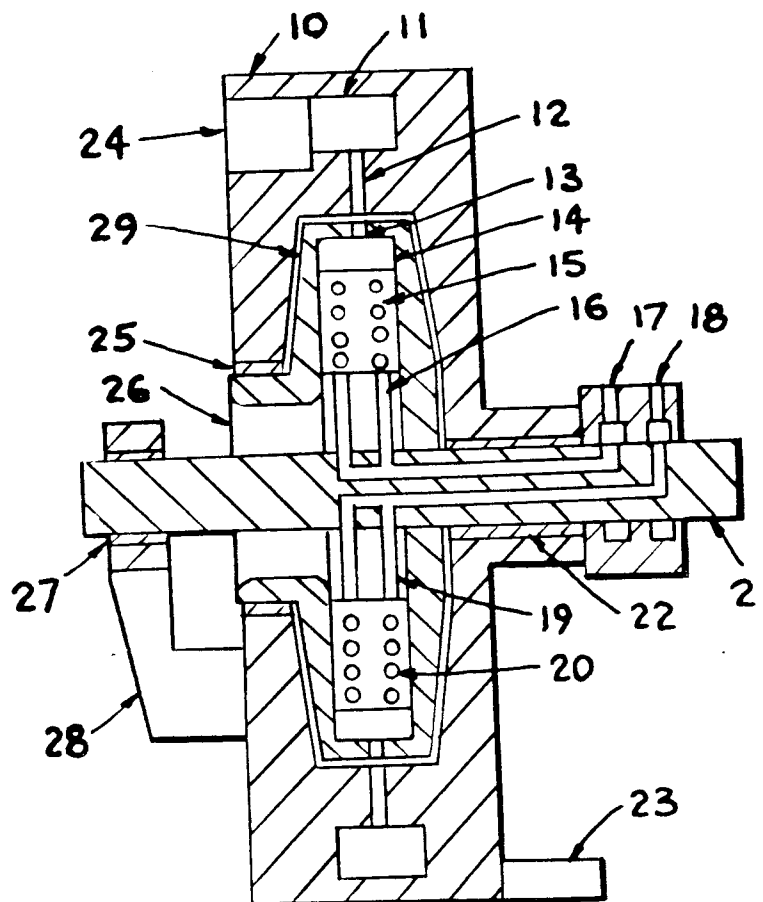
Primary Examiner—Henry F. Raduazo

Attorney, Agent, or Firm—Woffard, Felsman & Fails

[57] ABSTRACT

Method and apparatus for compressing gaseous fluids with minor amounts liquids, if any; by employing a rotating rotor wherein said fluid is compressed to a pressure that is normally higher than the fluid pressure immediately leaving said rotor; said fluid being usually further pressurized in a diffuser to utilize the kinetic energy of said fluid leaving at high velocity said rotating rotor; said rotating rotor being a centrifuge, with said fluid being compressed in the cavity of said rotor; said rotor cavity being provided with a cooling means to maintain nearly constant fluid temperature during said compression. Discharge nozzles from said rotor cavity are provided for said fluid near the periphery of said rotor; said nozzles may be arranged to discharge said fluid either radially, backward or forward as desired; said nozzles being either converging or diverging in shape as required to attain highest possible exit velocity for said fluid.

6 Claims, 3 Drawing Figures



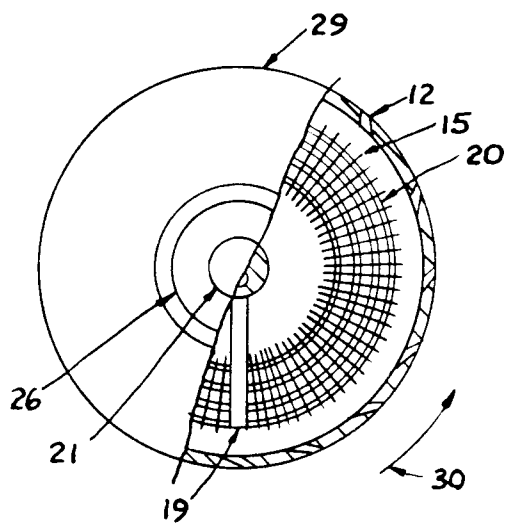


FIG 3

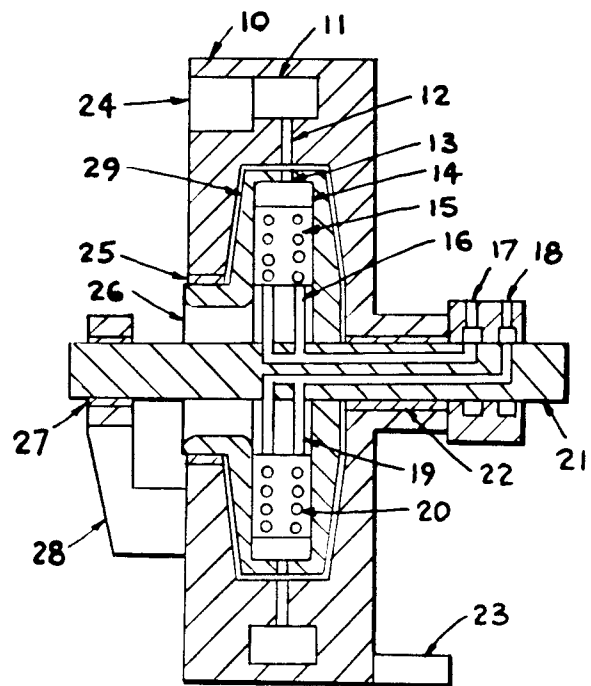


FIG 1

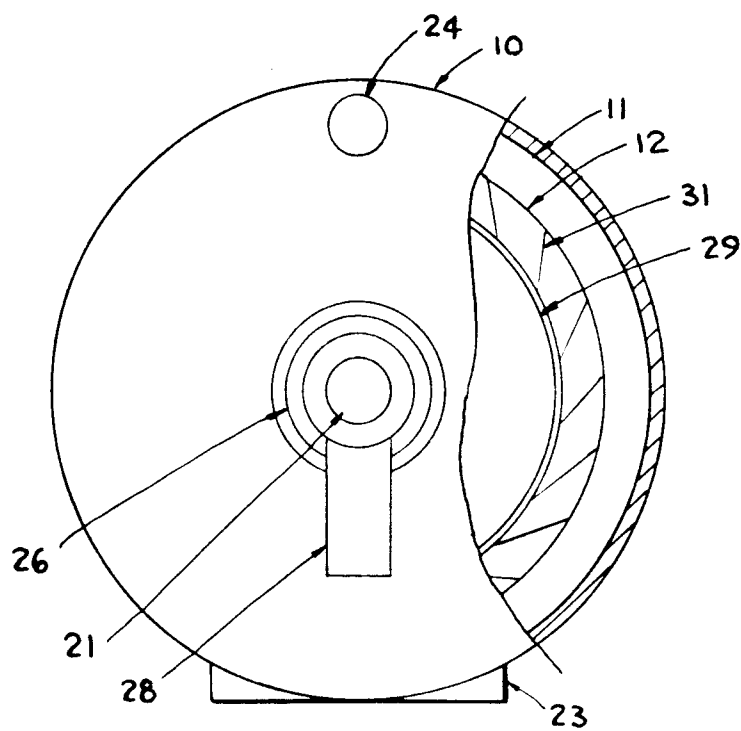


FIG 2

COMPRESSOR WITH COOLING

BACKGROUND OF THE INVENTION:

This invention relates generally to devices for compressing gases by employing centrifugal force to compress said gas.

The art of compressing gases has seen many devices. In some of those devices, a gas is accelerated within a rotating rotor passage and then discharged from said passage usually radially outward, and then said gas is compressed in a diffuser where the kinetic energy of said fluid is converted to pressure.

The main disadvantage of these conventional compressors is that their efficiency is rather poor due to friction in said diffuser due the very high velocities employed; losses when said kinetic energy is converted to pressure in said diffuser; and difficulty in obtaining higher pressures due to the very high rotor speeds needed.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 is a cross section of the compressor, and FIG. 2 is an end view of the same compressor.

In FIG. 3, the rotor is illustrated.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and apparatus for compressing gases wherein said gas is compressed within a rotating rotor cavity to a predetermined pressure; said compression being nearly isothermal, with the fluid temperature being maintained at a predetermined value by employing a cooling heat exchanger within said rotor. By cooling the fluid to be compressed during compression, the density of said fluid is increased, and the rotor speed needed to obtain a predetermined pressure increase is reduced.

Referring to FIG. 1, therein is illustrated a cross section of said compressor. 10 is casing, 11 is fluid collecting space, 12 is fluid diffuser passage, 13 is rotor discharge nozzle, 14 is fluid space within rotor, 15 is rotor heat exchanger fin and vane, 16 are coolant supply conduits to rotor heat exchanger, 17 is coolant supply and 18 is coolant return, 19 is coolant return conduit, 20 is coolant circulation conduit, 21 is rotor shaft, 22 is shaft bearing, 23 is compressor base, 24 is fluid exit opening, 25 is rotor seal, 26 is fluid inlet opening, 27 is shaft bearing, 28 is bearing support, 29 is rotor.

In FIG. 2, an end view of the same compressor shown in FIG. 1, is illustrated, with a section removed to show interior details. 10 is casing, 11 is fluid passage, 12 is diffuser passage with vanes 31, 29 is rotor, 24 is fluid exit opening, 26 is fluid inlet to rotor, 21 is rotor shaft, 28 is bearing support, 23 is compressor base.

In FIG. 3, an end view of the rotor is shown, with a section removed to illustrate interior details. 29 is rotor, 12 is fluid exit nozzle from said rotor, 15 are heat exchanger fins and vanes, 20 are coolant conduits, 26 is fluid inlet to rotor, 21 is shaft, 19 is coolant return conduit, 30 indicates rotation of rotor.

In operation, the fluid to be compressed enters said rotor via entry opening 26, and passes to rotor cavity. In said rotor cavity, said fluid is compressed by centrifugal action on said fluid by said rotor, with vanes 15 assuring that said fluid will rotate with said rotor. Cooling is provided within said rotor cavity to maintain a predetermined fluid temperature within said rotor cavity,

thus providing for nearly isothermal compression; or, in some cases, with suitable coolant temperature, having a reduction in fluid temperature within said rotor cavity, so that the fluid leaving said rotor has a lower temperature than the fluid entering said rotor. After said fluid has passed through said heat exchanger section, said fluid is discharged from said rotor via exit nozzles mounted near the periphery of said rotor; said exit nozzles being either converging or converging-diverging type as required to obtain desired exit velocity for said fluid. After leaving said rotor, said fluid will be passed to a stationary diffuser section, where kinetic energy of the fluid is converted to pressure. After leaving said diffuser, said fluid is passed to compressor discharge.

The fluid to be compressed may be a gas such as air, or be a mixture of gas and liquid. The coolant may be a liquid such as water, or be a gas with suitable properties so that the temperature increase for the coolant is less than for the fluid to be compressed. The fluid to be compressed, and the coolant fluid, are in heat exchange relationship within said rotor cavity, and with the temperature of the fluid to be compressed being higher than the temperature of said coolant fluid, heat transfer takes place from the fluid to be compressed to said coolant fluid.

The coolant fluid is supplied via the rotor shaft, and is returned to said rotor shaft. Little or no work is required for said coolant fluid, except due to fluid friction within said coolant conduits and passages.

To reduce fluid friction on said rotor exterior, said rotor may be closely fitted to said casing, as illustrated in FIG. 1. The rotating rotor will then partially evacuate the space between said rotor and said casing, with resultant reduction of drag on said rotor.

In FIG. 1, the compressor is shown to have a single fluid inlet. The said rotor may be provided with two inlets. Also, two or more stages may be provided in series, so that said fluid will be passed from one stage to next, with a gain in pressure in each stage.

Various well known devices, such as gauges, governors and the like, are employed with the device of this invention. They do not form a part of this invention and are not further described herein.

Said exit nozzles from said rotor may be arranged to discharge said fluid either forward, radially, or backward, as desired. In FIG. 3, said nozzles are indicated to be discharging in the forward direction.

What is claimed is:

1. A compressing centrifuge for compressing a fluid comprising:

- a casing for containing said fluid and for providing support for a rotor shaft and bearings; said casing having respective inlet and outlet ports for receiving said fluid to be compressed and for discharging the compressed fluid;
- a rotor shaft for power input needed to effect rotation of a rotor; said shaft being journaled for rotation in bearings supported in said casing; said first shaft having first and second longitudinally extending cooling fluid passageways for conveying a cooling fluid therethrough;
- a rotating centrifuge rotor for subjecting said fluid to a centrifugal force field; said centrifuge rotor being mounted on said shaft so as to rotate in unison with said shaft; said centrifuge rotor having an internal space with a plurality of heat conductive internal vanes defining respective cavities within

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- said primary rotor for transferring heat from a compressible fluid during centrifuge compression thereof and for ensuring that any fluid within said centrifuge rotor rotates with the same rotational speed as said centrifuge rotor; said centrifuge rotor being equipped with means for introducing the fluid to be compressed at the center of the centrifuge rotor and having suitable discharge opening adjacent the periphery for discharging the compressed said fluid; said discharge opening being smaller in cross sectional dimensions than the minimum cross sectional dimensions of the associated cavities upstream thereof for ensuring that the fluid within the respective cavities of the rotating primary rotor will be subjected to centrifugation and centrifugal compression for effecting at the periphery of the rotor and upstream of the discharge opening a compressed fluid having a second pressure that is higher than the pressure at the inlet to said centrifuge rotor;
- d. a cooling means disposed interiorly of said rotor so as to rotate in conjunction therewith; said cooling means comprising at least a peripherally disposed cooling passageway and respective radially extending passageways that are connected at their respective ends with said cooling passageway such that a cooling fluid can flow through said cooling means in heat exchange relationship with the compressed fluid in said rotor and back to said second passageway in said shaft for discharge of the heated cooling fluid; said cooling means having sufficient cooling surface to effect in conjunction with predetermined design conditions of inlet and outlet temperatures and flow rates of said cooling fluid substantially isothermal centrifuge compression of said fluid in said rotor;
- e. diffuser and fluid collecting section intermediate said rotor and said outlet port and communicating with both for converting a high velocity, cooled, compressed said fluid to a high pressure fluid upstream of said outlet port;
- f. a compressible first fluid being flowed through said inlet port, being substantially isothermally compressed within said rotor by being cooled within said rotor to a temperature less than the temperature would otherwise be if subjected to the same centrifuge compression without cooling, being discharged through said discharge opening at high velocity, having its pressure raised further in said diffuser and fluid collecting section and being passed out of said outlet port; and
- g. cooling fluid being flowed through said cooling means and said first and second cooling passageways in said shaft;
- whereby said fluid can be compressed to a predetermined pressure with a relatively low rotational speed for said rotor.

2. The compressing centrifuge of claim 1 wherein

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said casing is fitted so closely to the external walls of said rotor that centrifugal action on fluid particles will partially evacuate the space between said casing and said rotor walls to reduce the fluid friction and allow said rotor to rotate more freely for more efficient operation.

3. The compressing centrifuge of claim 1 wherein said discharge opening of said rotor comprises a plurality of discharge nozzles that are oriented to discharge the centrifugally compressed said fluid from said rotating rotor in the direction in which said rotor is rotating such that said predetermined pressure can be achieved with even lower said rotational speed.

4. The compressing centrifuge of claim 1 wherein said first fluid is air.

5. The compressing centrifuge of claim 1 wherein said cooling fluid is water.

6. A method of compressing a first gaseous fluid and simultaneously heating a second fluid comprising:

- a. subjecting said first fluid to a centrifugal force field via a centrifuge rotor having vanes defining cavities to ensure that the fluid attains the same rotational speed as said rotor, in a compressing centrifuge to compress the fluid to a first pressure that is higher at the periphery of said centrifuge rotor than at the entry thereto;
- b. cooling the compressed said fluid during its centrifuge compression to obtain substantially isothermal compression interiorly of said rotor by circulating said second fluid at an adjustable flow rate along the axis of rotation of said centrifuge rotor, radially outwardly and within said centrifuge rotor and adjacent the periphery of said centrifuge rotor in heat exchange relationship with the compressed said first fluid to heat said second fluid to a predetermined operational temperature commensurate with a predetermined resultant cooling of said first fluid during centrifuge compression; and passing the heated said second fluid radially inwardly to be discharged along the axis of rotation of said centrifuge rotor;
- c. passing said first fluid in its cooled, compressed state through discharge passageways that are smaller in cross sectional area than the minimum area of the respective associated cavities intermediate said vanes upstream of the discharge passageways to a lower second pressure and, thence, to a diffuser and fluid collecting section in which a large portion of the kinetic energy contained in the discharged fluid is converted to pressure to raise said pressure to a third pressure that is higher than said first pressure; and
- d. passing said fluid from said rotor to an outlet of said compressing centrifuge; the pressure of said fluid at said outlet of said compressing centrifuge being higher than at the inlet to said compressing centrifuge.

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[54] **HEAT AND STEAM GENERATOR**

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[21] Appl. No.: **376,877**

Related U.S. Application Data

[63] Continuation-in-part of Ser. Nos. 216,929, Jan. 11,
1972, Pat. No. 3,748,057, and Ser. No. 243,239,
April 12, 1972.

[52] U.S. Cl. **122/11, 122/26, 415/116**

[51] Int. Cl. **F22b 3/06**

[58] Field of Search **122/11, 26; 415/116, 178,
415/180**

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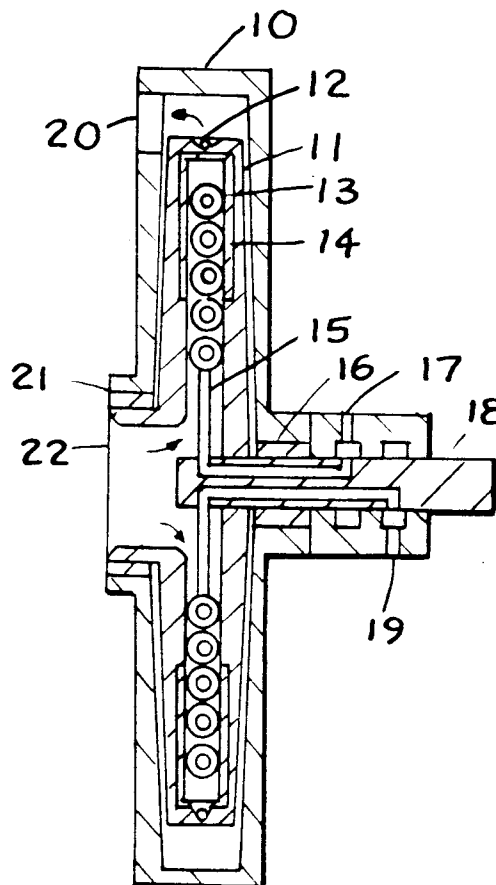
Primary Examiner—Kenneth W. Sprague

[57] **ABSTRACT**

A method and apparatus for the vaporizing of liquids

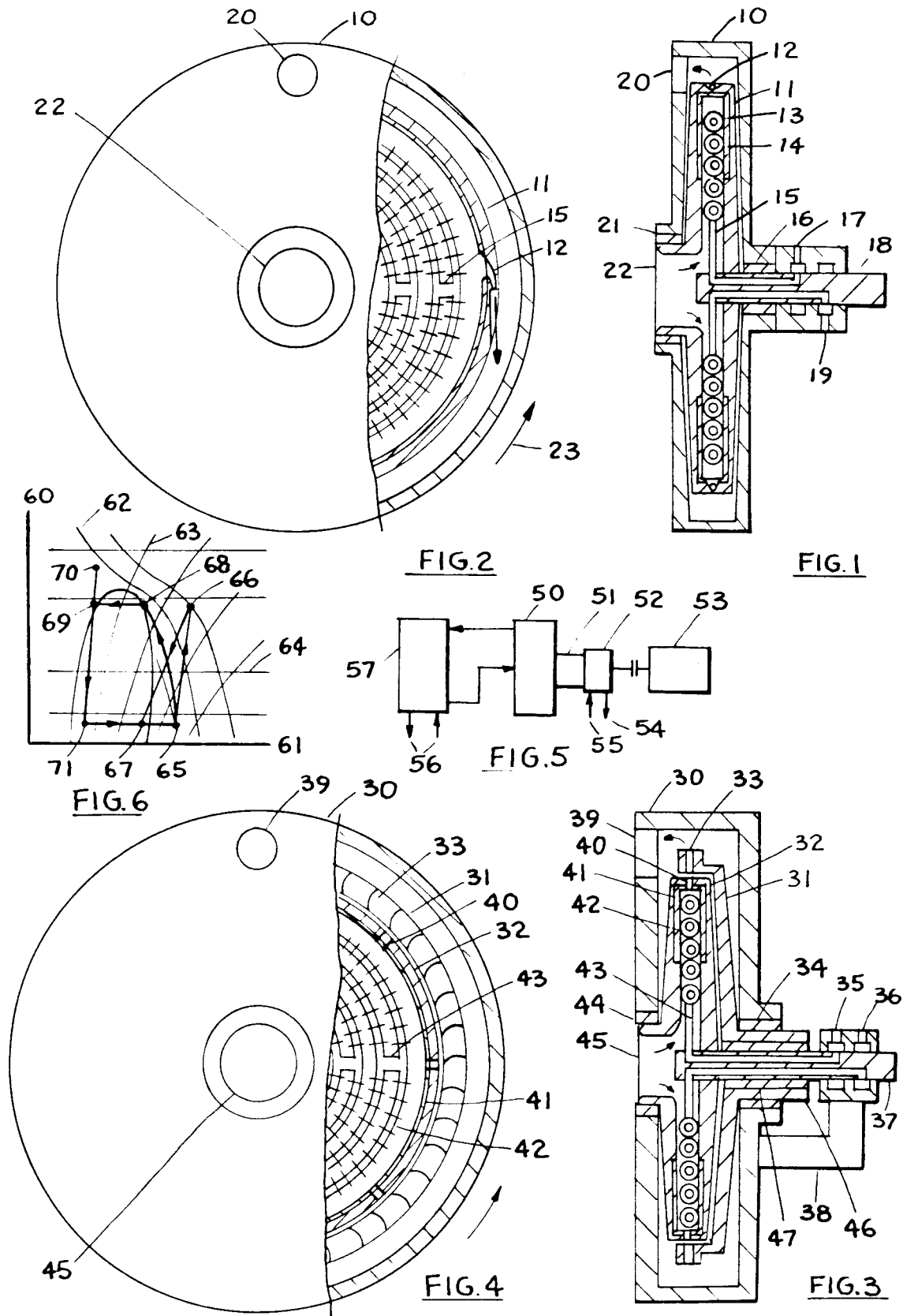
and for generation of steam, wherein a gaseous heating fluid is compressed with accompanying temperature increase, and this temperature increase is employed to provide for heat transfer to a working fluid which is in heat exchange relationship with said heating fluid during said compression. A rotating rotor with a heat exchanger mounted within is used to compress said heating fluid with said working fluid being within said heat exchanger conduits. Said gaseous heating fluid may also be condensed within said rotor with the heat of vaporization of said heating fluid being passed to said working fluid. Various fluids may be used for said heating fluid, such as halogenated hydrocarbons, ammonia, air, nitrogen or sulfur dioxide. For the working fluid, water, halogenated hydrocarbons, hydrocarbons or sulfur dioxide may be used. The fluids are so selected that said heating fluid has a higher temperature increase within said rotor than said working fluid so that heat transfer from said heating fluid to said working fluid may take place. By proper selection of the heating fluid, the work input of the unit rotor can be made to be nearly nil.

16 Claims, 6 Drawing Figures



PATENTED MAY 7 1974

3,809,017



HEAT AND STEAM GENERATOR

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part application of previous application titled "Rotary Compressor with Cooling," filed 1/11/72, Ser. No. 216,929, now U.S. Pat. No. 3,748,057. Claims relating to steam generation were deleted from said application at the request of the Examiner.

This is also a continuation-in-part of application titled "Compressing Centrifuge with Cooling," filed 4/12/72, Ser. No. 243,239.

BACKGROUND OF THE INVENTION

This invention relates generally to apparatus for generating steam and to apparatus for providing heating wherein a gaseous heating fluid is compressed with accompanying temperature increase; said temperature increase is then used to provide for heat transfer to a working fluid being in heat exchange relationship with each other within a rotating rotor.

The art of heating and steam generation has seen a variety of devices. In some of these devices, called boilers, heat is generated by burning a fuel; with said heat then being passed to a working fluid being circulated through said boiler.

These devices are inefficient since a large portion of the heat obtained from the fuel is passed to the stack with stack gases. Also, large amounts of costly fuel is required to generate the high temperatures required. Also, the burning of fuel results in pollution of the air.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the device, and FIG. 2 is an end view of the same device with a section removed to show interior details.

FIG. 3 is a cross section of another form of the device, and FIG. 4 is an end view of the unit shown in FIG. 3, with a portion removed to show interior details.

FIG. 5 is a schematic diagram showing typical installation of the device of this invention.

FIG. 6 is a pressure-enthalpy diagram for the heating fluid being circulated through the rotor.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and apparatus for the generation of heat and production of steam using low temperature heat sources to generate said steam. Said heat source may be ambient air, water at its natural temperature, or be a fluid supplied from another process or system. It is also an object of this invention to provide means for generating steam without air pollution.

Referring to FIG. 1, therein is illustrated a form of the device, shown in cross section. The heating fluid enters the rotor 11 via entry 22, and is passed to rotor cavity, where said rotating rotor 11 by centrifugal action will compress said heating fluid with accompanying temperature increase. During said compression of said heating fluid, heat is transferred from said hot heating fluid to working fluid being circulated within heat exchanger 13, and after compression, said heating fluid is discharged via exit nozzles 12 from said rotor; after this, said heating fluid leaves the unit casing space

around said rotor via opening 20. 10 is casing, 14 is thermal insulation within rotor to prevent overheating said rotor 11 wall which could weaken the wall material and cause failure; 15 is working fluid distribution tube; 16 and 21 are rotor bearings and seals, 17 is working fluid inlet to rotor shaft passage and 19 is working fluid outlet from rotor shaft passage, 18 is rotor shaft onto which said rotor is fixed.

In FIG. 2, an end view of the unit shown in FIG. 1, is illustrated. 10 is casing, 11 is rotor, 12 is rotor exit nozzle for said heating fluid, 15 is working fluid distribution conduit, 20 is heating fluid outlet, and 22 is heating fluid inlet. 23 indicates direction of rotation for said rotor 11.

In FIG. 3, another form of the device is shown, in a cross section. This device is provided with two rotors; power is supplied to the first rotor where the compression of the said heating fluid takes place and the heat transfer from said heating fluid to said working fluid occurs; the second rotor is used to convert the kinetic energy of the heating fluid leaving said first rotor, to work, and thus said second rotor will produce work. Usually the said first rotor and said second rotor are connected together through a gear box, so that the work produced by said second rotor is passed to said first rotor, thus reducing the required work input to said device. Heating fluid enters the rotating first rotor 32 via entry opening 45, and is compressed within said rotor 32 by centrifugal action on said fluid with accompanying temperature increase. Heat is transferred to said working fluid being circulated within heat exchanger 42, said working fluid being supplied via hollow rotor shaft 37 through entry 35, and through distribution conduit 43, and then returned through said rotor shaft to exit 36; said heating fluid being discharged from said rotor via nozzles 40 and passed to second rotor 31 vanes 33, where the kinetic energy contained in said high velocity heating fluid is converted to work; said second rotor being supported by shaft 46, and by bearings and seals 34. The first rotor shaft 37 is supported by bearing and seal 47, and by seal 44. Unit casing is 30, heating fluid outlet is 39, 41 is thermal insulation within rotor 32, and 38 is a support for working fluid connections.

In FIG. 4, an end view of the unit shown in FIG. 3, is illustrated. 39 is heating fluid outlet from casing 30, 33 is second rotor vanes, 31 is second rotor, 40 are first rotor exit nozzles for heating fluid, 32 is first rotor, 43 are distribution conduits for working fluid, 41 is a layer of thermal insulation, 42 is heat exchanger, 45 is heating fluid entry.

In FIG. 5, a typical schematic diagram is shown indicating connections for the heat generating device of this invention. 50 is the heat generating device. 51 is a gear box, 52 is a connector for supplying working fluid to the device, 53 is drive unit, 54 and 55 are supply and return for said working fluid, 57 is a heat exchanger for adding heat to said heating fluid, said heat exchanger 57 may be stationary; 56 are supply and return connections for a fluid to be used for adding heat to said heating fluid. Also, said heat exchanger 57 may be an air coil where air is used as the fluid for adding heat to said heating fluid.

In FIG. 6, a typical pressure-enthalpy diagram is shown for the heating fluid being circulated within said rotating rotor. 60 is pressure line, and 61 is an enthalpy line. 62 are constant temperature lines, 63 are constant

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entropy lines, 64 are constant pressure lines. For the unit shown in FIG. 1, the compression of the heating fluid is indicated by line 65-66, and the expansion of the heating fluid in the rotor exit nozzles is indicated by line 66-67. Heat is being added to the heating fluid from point 67 to point 65.

Also, in FIG. 6, typical cycle diagram is shown for the unit of FIG. 3. The compression within rotor 32 is indicated by line 65-68-69-70, and expansion in first rotor exit nozzles is indicated by line 70-71. Heat is then added to the heating fluid between points 71-65. There is no change in enthalpy in the second rotor vanes.

It should be noted that the compression within said rotating rotor is nearly non-flow, and thus the changes in the condition of the heating fluid can be shown approximately only on an enthalpy diagram during said compression. The expansion within the rotor exit nozzles is a continuous flow process, and the process can be shown on an enthalpy diagram. The radial velocity of the heating fluid within said rotor, is controlled by sizing the rotor exit nozzles to provide for a predetermined flow velocity; said radial flow velocity should be limited to values below 200 feet per second. Higher speeds for the heating fluid radially, will tend to reduce the pressure near the rotor periphery, and thus also reduce the heating fluid temperature.

The rotor exit nozzles for the form of the device shown in FIG. 1, are arranged to discharge the heating fluid in backward direction, preferably tangentially backward. The nozzles are sized and shaped to provide for maximum attainable exit velocity for the fluid leaving said nozzle, relative to said nozzles. The first rotor exit nozzles for the unit shown in FIG. 3, may be arranged to discharge said heating fluid radially, backward or forward, as desired; said nozzles are sized and shaped to provide for highest attainable exit velocity for said heating fluid from said first rotor nozzles.

The unit shown in FIG. 3, is mainly intended to be used when the said heating fluid is being condensed within said first rotor, with the heat of vaporization of said heating fluid being passed to said working fluid. Usually the exit velocity of the said heating fluid leaving said first rotor nozzles in liquid form is low, and the use of said second rotor allows converting the kinetic energy of said heating fluid to work, thus reducing the work input to said heat generating unit. The unit shown in FIG. 3 may also be used when said heating fluid remains a gas. The unit shown in FIG. 1 is mainly intended for use when the heating fluid remains a gas when in rotor; said unit may also be used when the heating fluid is condensed, within said rotor. The main advantage of providing for condensation of said heating fluid within said rotor, is in the amount of heat transferred from said heating fluid to said working fluid, which is much larger. Thus, the size of heating fluid passages for a predetermined amount of heat transfer, is much smaller in a unit where the heating fluid is being condensed within rotor than in a unit where said heating fluid remains a gas.

In operation, power is supplied to the rotor shaft, to provide for rotating said shaft. In the unit of FIG. 1, the heating fluid leaving said rotor nozzles, produces thrust, which will reduce the power required to rotate said rotor shaft. In the unit of FIG. 3, work is produced in said second rotor, which can be used to reduce work input to said unit.

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There are numerous applications for this heat and steam generator. The units can be used to provide heat as needed for heating buildings, heating liquids, and producing steam; the heat source normally being at a much lower temperature.

The fluids being used with the devices of this invention, are a gaseous fluid as the heating fluid which may be condensed and leave the unit as a liquid; and a normally liquid working fluid, which may be vaporized and turned to steam. Said steam may be formed within said heat exchanger of said rotor, or may be formed in a flash tank after said working fluid leaves said rotor. Further, both fluids may be gases, where the two fluids are so selected that the temperature increase for said heating fluid is greater than for said working fluid, within the centrifugal force field of said rotor, so that heat transfer may take place from said heating fluid to said working fluid.

The work input to said rotor is low, since the heating fluid provides during 147 its exit from the rotor, thrust to help rotate said rotor, in the form of FIG. 1. The working fluid enters and leaves said rotor via shaft, and thus the work input for said working fluid is nearly nil, within said rotor. Thus, the work input required to rotate the rotor of the unit of FIG. 1, when the heating fluid is properly selected, and the amount of heat removed is properly controlled, may approach zero; this is due to pressure increase of the the said heating fluid during compression within rotor due to cooling of said heating fluid during said compression; this increased pressure will normally also result in increased exit velocity for said heating fluid from said rotor exit nozzles. Therefore, the device of this invention may be also used to provide steam or pressurized vapor, as required for power generation, while using low temperature heat sources to generate said steam or vapor.

Further, it is possible, by proper selection of the heating fluid, to provide a unit capable of producing high pressure water steam as required by modern steam turbines in power plants. Such heating fluids include many of the halogenated hydrocarbons, such as Freon 12, Freon 13B1, and others; also fluids such as Bromine. For these applications, said heating fluid must have a low ratio of specific heat at constant pressure divided by specific heat at constant volume, and also low value of specific heat at constant pressure; above mentioned fluids meet these requirements. Since the maximum tangential speed for most materials of construction of the rotor is about 1650 feet per second, the use of the types of fluids mentioned above as said heating fluid is required, to provide for sufficiently high temperatures to provide water steam to modern power turbines.

The thermal insulation layer 14 in FIG. 1, is provided to prevent excessive heating of the rotor wall near the rotor periphery where the heating fluid temperature is high. The heat exchanger is in contact with the rotor wall near the shaft, thus providing cooling for said wall, and helping to maintain rotor strength; said working fluid being relatively cold near the shaft, and also said heating fluid being cooler near said shaft.

The heat exchanger within the rotor is shown to be constructed from finned tubing, with the heating fluid passing through the fins; also, the fins provide needed obstructions so that said heating fluid will rotate with said rotor. Other types of heat exchanger construction may be used to obtain the same effect. The working fluid is arranged normally to be in parallel flow with

said heating fluid, with said working fluid entering said heat exchanger nearest to the shaft, and then progressing outward toward rotor periphery; then passing back to the said shaft and from there to exit. This type flow pattern is necessary to obtain highest working fluid temperatures. For heating water, for example, any suitable flow pattern may be employed, since the final temperature is not very high.

The rotor sides may be closely fitted to the unit casing to provide for reduced fluid friction on said rotor at high speeds. Alternate methods, such as friction discs, may be also employed to reduce said fluid friction, on outside surfaces of the rotor.

The rotor may be constructed of any suitable material; high strength steel would be normal. The rotor walls are made thick at center, and taper toward periphery to provide for necessary strength for high speeds, as shown.

The work input to the rotor to rotate said rotor, is composed of the work required to accelerate said heating fluid, and said working fluid to rotor speed, to overcome friction on bearings and seals, to overcome fluid friction on rotor external surfaces. Since the working fluid is returned to the rotor center, work is reclaimed from said working fluid, and the work demand by working fluid is nil. The heating fluid leaves the rotor at a high velocity producing thrust on the rotor nozzles, thus producing work; therefore, by proper selection of said heating fluid, and with controlled heat removal from said heating fluid, the work output by the heating fluid can be made to approach the work input for said heating fluid to accelerate said fluid; also, in some situations, the work output by the heating fluid may exceed the work input required for said heating fluid. The fluid friction on the outer surfaces of the rotor can be reduced by fitting closely the rotor walls to the casing as illustrated in FIG. 1, and by centrifugal action the space between the rotor and casing can be evacuated thus reducing the friction loss. The friction loss in bearings and seals is normally very low, when expressed per pound of steam generated. Thus, the work input to generate a pound of steam by using this device, can approach very low value, and can be considered to be negligible, while a low temperature heat source is used to make said steam. Also, it should be noted that the work demand by this device is independent of the amount of heat transferred, as described herein. Further, by proper selection of the heating fluid, and by arranging the heat transfer amount, the work output by the heating fluid can be made larger than the total work input to the rotor so that steam generator accessories may be driven by said rotor; these accessories including for example feed water pump, and other items as needed.

What is claimed is:

1. A heat and steam generator comprising:

- a. a rotor for generating said heat and steam by compressing a compressible heating fluid within said rotor by centrifugal action on said fluid by said rotating rotor with accompanying temperature increase of said heating fluid; said rotor having an entry opening for said heating fluid near the center of rotation; said rotor having a cavity to contain said heating fluid during said compression, and having exit nozzles mounted near the rotor periphery for discharging said heating fluid after said heating fluid has been compressed within said ro-

tor; said rotor having vanes or other obstructions within said rotor cavity to assure that the said heating fluid will rotate with said rotor with approximately the same tangential velocity as said rotor; said rotor having a heat exchanger within said rotor cavity; within said heat exchanger a working fluid is being circulated in heat exchange relationship with said heating fluid; said working fluid being supplied to said rotor heat exchanger via suitable passages within the rotor shaft and returned back to passages within said shaft; said rotor being fixed to said rotor shaft and said shaft being rotatably mounted on bearings; power being supplied from an external source to said rotor shaft at least initially to rotate said shaft; said working fluid receiving heat from said heating fluid within said rotor;

- b. a casing to support said rotor and its shaft;
- c. a heating fluid being circulated within said rotor and being a compressible fluid and being a gas at least initially when entering said rotor;
- d. a working fluid being circulated through said rotor heat exchanger and being a fluid that is less compressible than said heating fluid with lower temperature increase within said rotor than for said heating fluid.

2. The device of claim 1 wherein said rotor exit nozzles are arranged to discharge tangentially backward and said nozzles are sized and shaped to provide for highest attainable exit velocity from said nozzles for said heating fluid, for the pressure differential between the entry and exit ends of said nozzles.

3. The device of claim 1 wherein said rotor exit nozzles arranged to discharge said heating fluid at highest attainable exit velocity for the pressure differential available between the entry and exit ends of said nozzles; and wherein a secondary rotor is provided with suitable vanes to convert the kinetic energy available in the heating fluid stream to power; with said secondary rotor being supported by shaft and bearings and being rotatably mounted and with means for passing said power to external load.

4. The device of claim 1 wherein said heating fluid radial velocity within said rotor cavity is maintained at less than 200 feet per second.

5. The device of claim 1 wherein said heating fluid enters said rotor as a gas, and wherein said heating fluid is condensed within said rotor cavity with the heat of vaporization of said heating fluid being passed to said working fluid being circulated within said heat exchanger.

6. The device of claim 1 wherein said heating fluid enters said rotor as a gas and wherein said heating fluid leaves said rotor nozzles as a gas, with minor amounts of liquid, if any.

7. The device of claim 1 wherein said heating fluid is a gaseous fluid with a low specific heat at constant pressure; with said specific heat being less than 1.2 British thermal units per pound.

8. The device of claim 1 wherein said heating fluid a halogenated hydrocarbon.

9. The device of claim 1 wherein said working fluid is water.

10. The device of claim 1 wherein said working fluid is a halogenated hydrocarbon.

11. The device of claim 1 wherein said rotor is provided with thermal insulation within said rotor near the

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periphery of said rotor, and wherein said rotor walls are in contact with said heating fluid in the areas near the rotor center; the thermal insulation to reduce the loss of strength of said rotor construction material due to high temperatures.

12. The device of claim 1 wherein said heating fluid is supplied to said rotor from an external stationary heat exchanger, and is also passed from said device back to said external heat exchanger for the purpose of adding heat to said heating fluid, with said heating fluid being circulated in a closed loop.

13. The device of claim 1 wherein said heating fluid is ambient air.

14. The device of claim 1 wherein said heating fluid is provided with means, external to said device, for adding heat from water, at its natural temperature, to said heating fluid, with said heating fluid being circulated

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through said means.

15. The device of claim 1 wherein said flow of said working fluid is arranged within said heat exchanger to be outward from the center of said rotor toward rotor periphery thus having the said working fluid in parallel flow with said heating fluid during the period when heat is transferred from said heating fluid to said working fluid; said working fluid then being returned back to said rotor center directly from said rotor periphery.

16. The device of claim 1 wherein said heating fluid and the amount of heat removed from said heating fluid is selected to provide for a sufficiently high exit velocity from said rotor exit nozzles to generate sufficient work to overcome all rotor work losses and to make the work input to said rotor shaft during normal operation nil.

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[54] **TURBINE**

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[22] Filed: **Oct. 9, 1973**

[21] Appl. No.: **404,406**

[52] U.S. Cl. **415/81, 415/178**

[51] Int. Cl. **F01d 1/16, F01d 5/08, F01d 25/08**

[58] Field of Search 415/80, 81, 178, 179, 63,
415/82; 60/39.35

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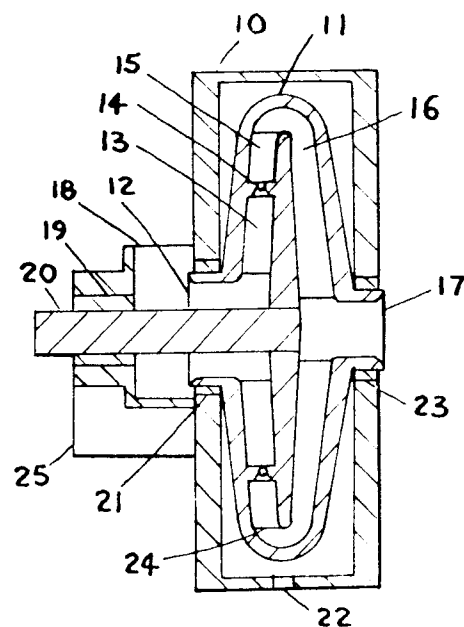
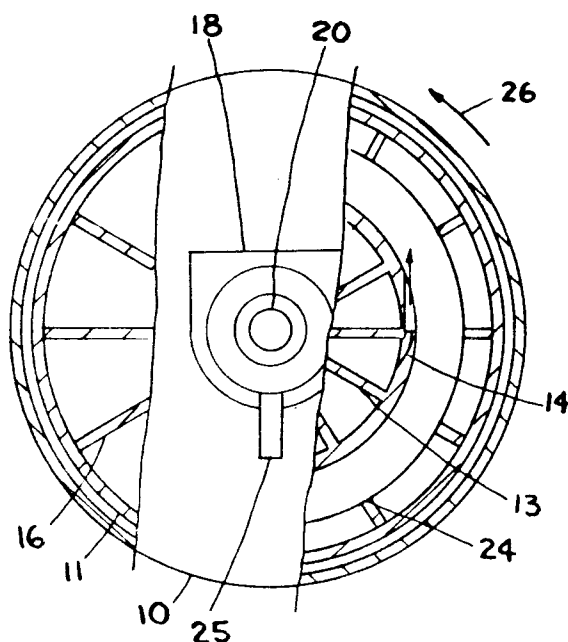
Primary Examiner—C. J. Husar

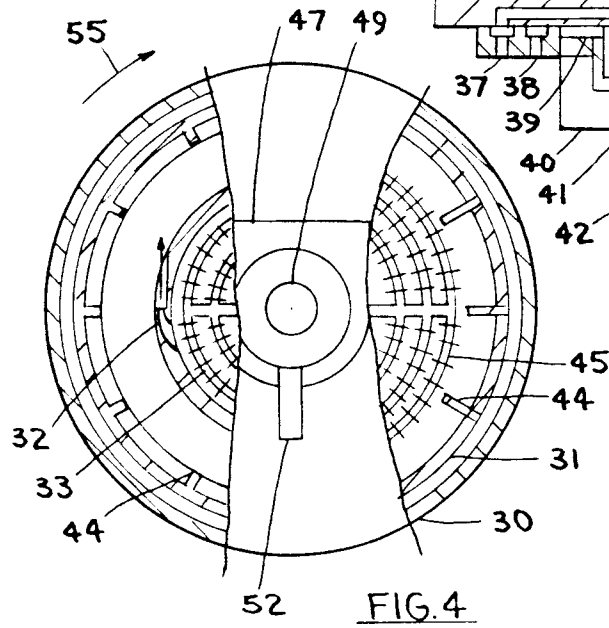
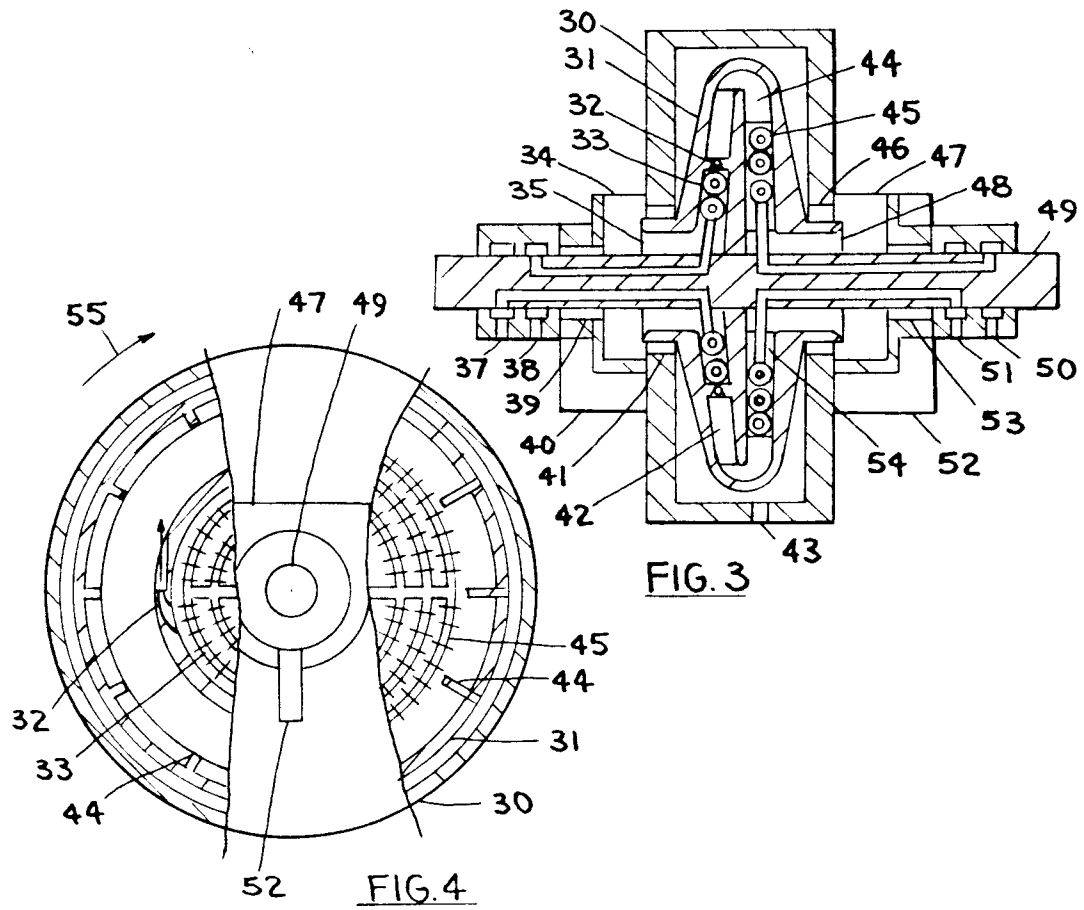
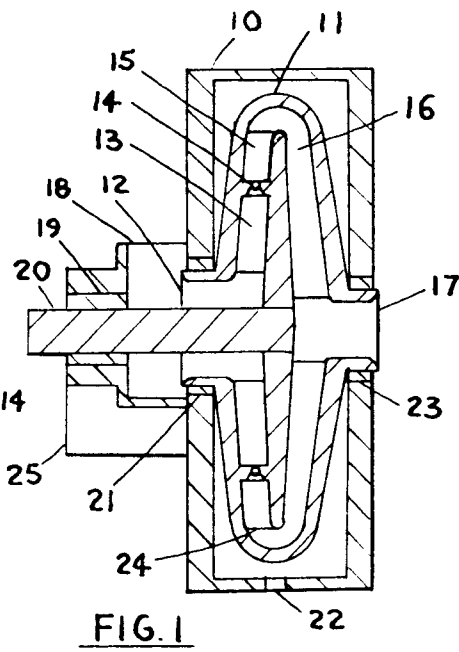
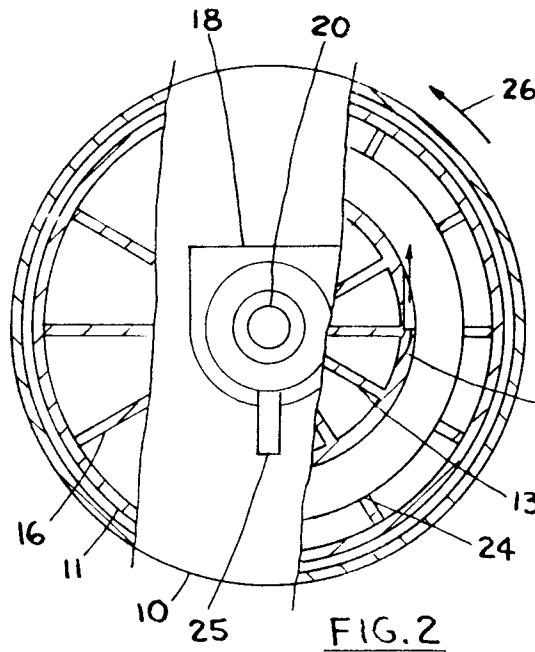
Assistant Examiner—L. J. Casaregola

[57] **ABSTRACT**

A method and apparatus for generating power by passing a fluid from a higher energy level to a lower energy level through a rotating turbine wheel. The fluid is first compressed within said turbine rotor, with accompanying pressure increase, and the said fluid is accelerated within said rotor, thus obtaining a fluid that has a high absolute velocity and elevated pressure, this fluid is then passed to a second portion of the rotor where the kinetic energy contained by said fluid is converted to work. This work is then passed to the rotor shaft as the work output of the turbine. In an alternate arrangement, heat may be added to the fluid during its passage through the rotor increasing the work output of the turbine.

6 Claims, 4 Drawing Figures





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TURBINE

BACKGROUND OF THE INVENTION

This invention relates generally to devices for generating power in response of a fluid being flowed from a higher energy level to a lower energy level by passing said fluid through a rotating turbine rotor.

There have been various types of turbines previously; in some of these a fluid is accelerated in a single or multiple stationary nozzles and then passed to vanes mounted on a rotating rotor wheel, where the kinetic energy of the moving fluid is converted to power.

These conventional turbines normally have high energy losses due to fluid friction, especially between the rotor vanes and the fluid where the velocity differentials are usually large. Also, these turbines often require complex shaped turbine vanes making the unit costly.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the turbine, and

FIG. 2 is an end view of the same unit.

FIG. 3 is a cross section of another form of the turbine, and

FIG. 4 is an end view of the unit of FIG. 3.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a turbine for generating power which is simple in construction and low in cost, and which has a high efficiency of operation with low fluid velocities relative to the turbine rotor. Further, it is an object of this invention to provide a turbine where the turbine rotor may be encased within a vacuum casing to reduce friction losses on rotor external surfaces. Also, it is an object of this invention to provide a turbine wherein the fluid passing through the said turbine rotor may have heat added to it from another heating fluid being circulated in heat exchange relationship with said fluid either during expansion, during compression, or during both compression and expansion, for increased power output from said turbine.

Referring to FIG. 1, therein is illustrated a form of the turbine in cross section; and without said heat exchanger. 10 is casing, 11 is rotor. Fluid enters via entry opening 18 and passes to rotor 11 via opening 12, and from there to rotor compression cavity where vanes 13 will assure that the fluid will rotate with said rotor; after compression and pressurization, said fluid is passed through nozzles 14 oriented to discharge said fluid forward so that the absolute tangential velocity of said fluid will be the sum of the tangential velocity of said nozzles 14 and the discharge tangential velocity of the fluid. Said fluid enters a space 15 within said rotor, and then passes to the expansion side of the rotor where the motive energy of the fluid is converted to work, with vanes 16 assuring that said fluid will rotate with said rotor. The fluid then leaves via exit opening 17. 20 is rotor shaft fixed to rotor 11, 19 is rotor shaft bearing, 25 is bearing support, 21 and 23 are rotor seals, 22 is vent communicating with space within casing and around rotor, and 24 indicates the point where fluid enters rotor area with vanes to control fluid flow with these vanes extending to the rotor exit near 17.

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In FIG. 2, an end view of the turbine is illustrated, with some sections removed to show internal details. 10 is casing, 11 is rotor, 16 are vanes for converting motive energy to work, 18 is fluid inlet, 20 is rotor shaft, 26 indicates direction of rotor rotation, 14 is fluid nozzle mounted on a divider within rotor cavity, adapted for passing through said fluid, 13 are vanes within rotor cavity, 24 are points where fluid enters expansion side vanes, 25 is bearing support.

In FIG. 3, a form of the turbine is shown where heat exchangers are placed within rotor cavity to add heat to the fluid; the turbine is shown in cross section. 34 is fluid entry, 35 is fluid entry to rotor, 33 is heat exchanger in rotor cavity, compression side; 32 are nozzles, 42 is space into which fluid enters on leaving said nozzles, 44 are rotor vanes, 45 is another heat exchanger within the expansion section of the rotor cavity, 48 is rotor fluid exit, 47 is fluid exit, 30 is casing, 31 is rotor, 41 and 46 are rotor seals, 49 is rotor shaft, 50 and 51 are heating fluid inlet and outlet to heat exchanger 45, 52 is bearing support, 53 and 39 are shaft bearings, 37 and 38 are heating fluid inlet and outlet to heat exchanger 33, 40 is bearing support, 43 is casing vent into which a vacuum pump may be connected to evacuate the space between rotor and casing.

In FIG. 4, an end view of the unit shown in FIG. 3 is illustrated. 47 is fluid exit, 49 is rotor shaft, 45 is heat exchanger, 44 is rotor vane, 31 is rotor, 30 is casing, 52 is bearing support, 44 is rotor vane, 33 is heat exchanger, 32 is fluid nozzle mounted on divider wall, 55 indicates direction of rotation for rotor.

In operation, fluid at higher energy level enters rotating rotor 11 via entry 12 and is pressurized within said rotor by centrifugal force acting on the fluid, with vanes 13 assuring that the fluid will rotate with said rotor. After compression, the fluid is passed through nozzles 14 in forward direction so that the tangential velocity of the fluid leaving said nozzles is added to the tangential velocity of the said rotor. Thus said fluid will have an absolute tangential velocity on discharge side of said nozzles that is greater than the tangential velocity of said rotor in the same area. The fluid will then enter to the expansion side of the rotor where the motive energy contained by said fluid is converted to work. The tangential fluid velocity and the tangential rotor velocity are usually the same at point 24 where the exit side vanes start, so as to provide for smooth passage of the fluid to spaces between said vanes 16, the velocity of the said fluid may be different at point 24 than the rotor velocity, and the vanes may be suitably shaped to provide for smooth passage for the fluid between said vanes. After passing between said vanes, said fluid is decelerated as it passes toward the rotor center and work is thus obtained from said fluid, this work is then passed to the rotor shaft 20. Work is required to accelerate said fluid when passing through vane spaces on inlet side, and through nozzles 14; the net work output for this turbine is the work put out by vanes 16 less the acceleration work in vanes 13 and in nozzles 14. After said fluid has been decelerated in vanes 16, said fluid is discharged via exit 17 at a lower energy level than at entry 12.

The operation of the unit shown in FIG. 3, is similar to that described hereinbefore for the unit of FIG. 1, except that heat may be added to the working fluid in the heat exchangers. Either one of the heat exchangers 33 and 45 may be used, or both may be used simulta-

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neously. Normally, the radial velocity of the fluid through the turbine rotor is low, and heat addition to the working fluid may be readily provided. Heat addition during compression by heat exchanger 33 increases the enthalpy of the working fluid, with some reduction in density as compared to a unit without a heat exchanger. This increase in enthalpy and a reduction in density will allow a greater tangential tip velocity for vanes 44, thus resulting in increased work output. The addition of heat during expansion in heat exchanger 45, will reduce the density of the fluid being expanded and this in turn will reduce the pressure on the exit side of the rotor where vane 44 is located, thus allowing a greater tip velocity for vanes 44 and resulting in greater work output. The tip velocity for vanes 44 being the tangential velocity of said vanes near the periphery of said rotor.

It should be noted that normally the working fluid velocity leaving the nozzles 32 and nozzles 14 is relatively low, and is normally below sonic velocity even when gaseous fluids are used as said working fluid. Said nozzles 32 and 14 are sized and shaped for the fluid, to provide for highest attainable exit velocity for the fluid for the available pressure and enthalpy differential between the entry and exit ends of said nozzles.

The working fluid is normally either a gas or a liquid. Fluids containing both gaseous and liquid components can not usually be passed through this turbine; minor amounts of liquids may be discharged from the rotor via small vents that may be provided near the rotor periphery. Such vents are not shown on the drawings.

The heating fluid may either a gas or a liquid. The heat exchanger passages for the heating fluid are indicated in the drawings to be relatively small, and more suited for liquids. Gaseous heating fluids may be used if said heating fluid passages are suitably enlarged. Also, additional heating can be obtained with gaseous heating fluids by proper selection of said fluids so that advantage may be taken from the heat of compression of such heating fluid within said heat exchangers when such heating fluid is compressed within the heat exchanger coils with accompanying temperature increase.

The rotor in the drawings is shown shaped to allow for high rotational speeds, with heavy material sections especially near the rotor center. The heat exchangers are shown having been made of finned tubing; other types of heat exchanger arrangements may be used. Thermal insulation may be provided to prevent undesirable heat transfer within rotor; such insulation has not been shown as it is within existing art.

The turbine of this invention may be also made as a multi-stage unit, with the working fluid passing from one stage to next, with a reduction in energy level in each stage for the fluid. In these arrangements, two or more of the said rotor wheels are mounted on a single shaft, with the exit of a previous stage connected to the entry of the succeeding stage.

Various controls, gauges and governors are used with the turbine of this invention. These do not form a part of this invention and are not further described herein.

What is claimed is:

1. A turbine for generating power comprising:

- a. a casing for supporting a shaft;
- b. a shaft journaled in bearings for rotation and supported by said casing;
- c. a rotating rotor mounted on said shaft so as to rotate in unison therewith; said rotor having an entry

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port disposed near the center of said rotor for entry of a working fluid with said entry port communicating with radially outward extending passageways for passing said working fluid; said outward extending passageways having vanes therewithin for ensuring that said working fluid will rotate with said rotor for acceleration and for pressurizing said working fluid; said radially extending passageways being provided with nozzles at their outward ends for passing said working fluid with said nozzles being oriented to discharge said working fluid forward in the direction of rotation thus increasing the absolute tangential velocity of said working fluid to a value greater than the tangential velocity of said nozzles; said nozzles discharging said working fluid to a first peripheral cavity of said rotor; said working fluid being then passed from said first peripheral cavity to rotor inward extending passageways with said inward extending passageways being provided with vanes for assuring that said working fluid will rotate with said rotor and for generation of power associated with the deceleration of said working fluid; the outward ends of said vanes within said inward extending working fluid passageways being at radial distance greater from rotor center than the discharge ends of said nozzles; said inward extending passageways being connected with an exit port being disposed near the center of said rotor for discharge of said working fluid;

d. a fluid being flowed through said rotor; said fluid being flowed into said inlet of said rotor and being pressurized by centrifugal compression therewithin and discharged outwardly and substantially tangentially in said forward direction at a tangential velocity that is substantially higher than said tangential velocity of said nozzles; said fluid having high said tangential velocity because of the additive effects of the pressure velocity of the fluid under the differential pressure existing between the entry and exit ends of said nozzles plus the tangential velocity due to the tangential velocity of said nozzles.

2. The turbine of claim 1 wherein said working fluid is at a higher energy level at the said entry to said rotor than at said exit from said rotor.

3. The turbine of claim 1 wherein a heating heat exchanger is placed within said radially outward extending working fluid passageways for adding heat to said working fluid during and after compression and acceleration of said working fluid, with a heating fluid being circulated through said heat exchanger with said heating fluid being supplied and then returned via passages near the center of rotation of said rotor.

4. The turbine of claim 1 wherein a heating heat exchanger is placed within said radially inward extending working fluid passageways for adding heat to said working fluid during expansion and deceleration of said working fluid for the purpose of reducing the density of said working fluid; said heating fluid being circulated within a heat addition heat exchanger and supplied and returned via passages near the center of rotation of said rotor.

5. The turbine of claim 1 wherein said working fluid is a liquid.

6. The turbine of claim 1 wherein said working fluid is a gas.

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 [76] Inventor: Michael Eskeli, 7994-41 Locke Lee, 3,828,573 8/1974 Eskeli 62/401
 Houston, Tex. 77042 3,834,179 9/1974 Eskeli 62/401

[22] Filed: Jan. 31, 1974

Primary Examiner—Henry F. Raduazo

[21] Appl. No.: 438,239

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 410,985, Oct. 30, 1973.

[52] U.S. Cl. 62/401, 415/178, 165/86

[51] Int. Cl. F25d 9/00, F28d 11/00

[58] Field of Search 410/1; 415/178, 179; 165/86; 62/401, 402, 499, 500

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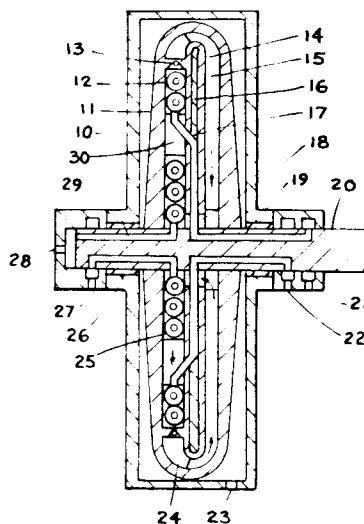
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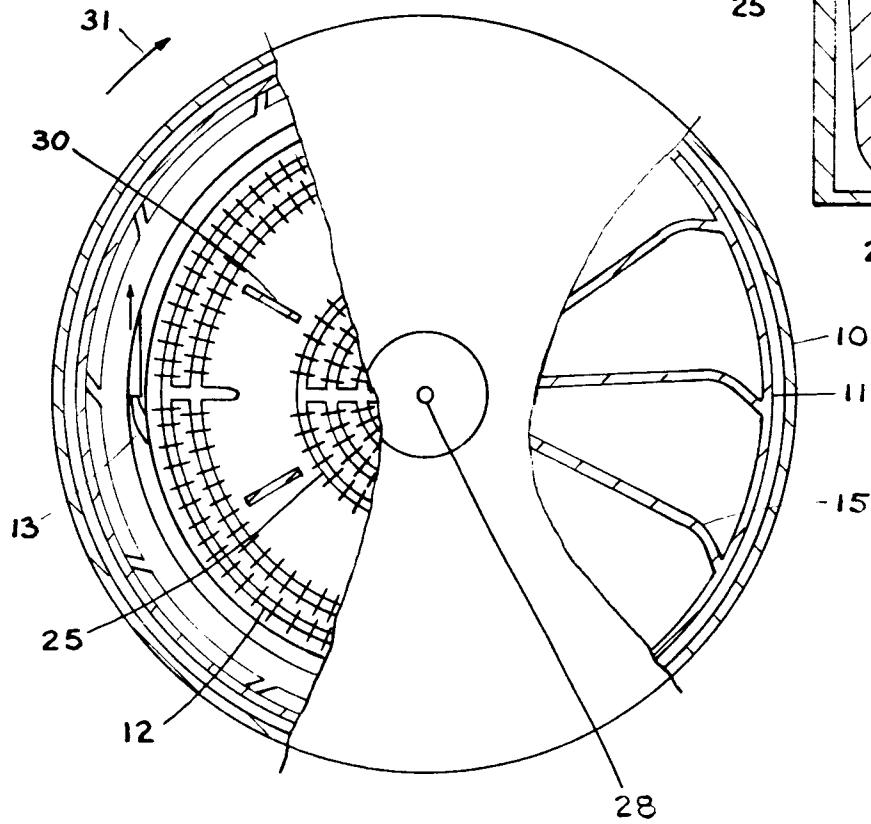
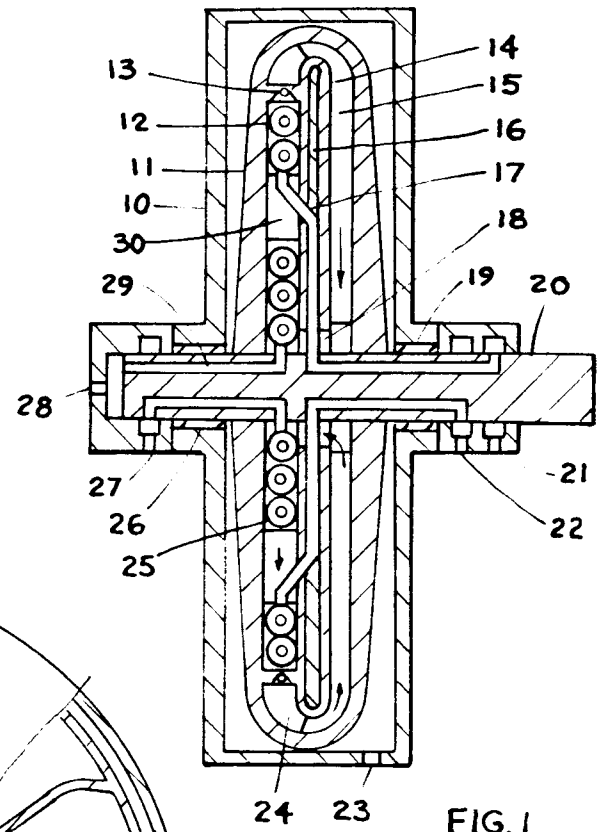
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[57] ABSTRACT

A method and apparatus for generating power wherein a fluid is accelerated and compressed within a rotating rotor in outward extending fluid passageways with removal of heat from said fluid during first part of said compression, and heat addition to said fluid during latter part of said compression, and wherein said fluid is then decelerated in passages extending inward with work being obtained during said deceleration. The fluid being used is normally a gaseous fluid, and is normally sealed and circulated within said rotor. The heating and cooling are provided by circulating a heating and cooling fluid through respective heat exchangers. Heat may be also added in part during said deceleration, and heat may be removed in part during said deceleration.

4 Claims, 2 Drawing Figures





SEALED SINGLE ROTOR TURBINE

CROSS REFERENCES TO RELATED APPLICATIONS

This is a continuation-in-part application of a previously filed application titled "Sealed Single Rotor Turbine," filed 10/30/73, Ser. No. 410,985.

This invention relates generally to devices for generating power in response of a fluid being flowed from a higher energy level to a lower energy level by passing said fluid through a rotating turbine rotor.

There have been various types of turbines previously; in some of these a fluid is accelerated in single or multiple stationary nozzles and then passed to vanes mounted on a rotating rotor wheel, where the kinetic energy of the moving fluid is converted to power.

These conventional turbines normally have high energy losses due to fluid friction, especially between rotor vanes and the fluid where the velocity differentials are usually large.

FIG. 1 is a cross section of the turbine, and

FIG. 2 is an end view of the same unit.

It is an object of this invention to provide a turbine where the working first fluid is sealed within the turbine rotor and heat is added to said first fluid from a second fluid during and after compression, and cooling is provided for said first fluid before and during first part of said compression; this being done to improve the performance of said turbine.

Referring to FIG. 1, therein is shown a cross section of the turbine. 10 is casing, 11 is rotor, 12 is heat addition heat exchanger, 13 are rotor nozzles for first fluid, 14 is rotor dividing wall, 15 is a vane within inward first fluid passages, 16 is a layer of thermal insulation, 17 is second fluid distribution conduit, 18 is first fluid passage near rotor center, 19 and 26 are rotor bearings and seals, 20 is rotor shaft, 21 and 22 are second fluid inlet and outlet, 23 is casing vent into which a vacuum source may be connected, 24 is first fluid space within rotor, 25 is heat removal heat exchanger, 27 and 28 are third fluid inlet and outlet, 29 is third fluid passage within shaft 20, 30 is vane within outward extending first fluid passages.

In FIG. 2, an end view of the unit shown in FIG. 1, is illustrated with portions removed to show internal details. 10 is casing, 11 is rotor, 15 is vane within inward extending first fluid passages, 28 is third fluid outlet, 12 is heat addition heat exchanger, 25 is heat removal heat exchanger, 13 is rotor nozzle, 30 is vane, and 31 indicates direction of rotation for rotor.

In operation, first fluid enters the outward extending first fluid passages via opening 18, and is accelerated and compressed as it passes outward by centrifugal action on the fluid by said rotating rotor. During first part of said compression, heat is removed from said first fluid in heat exchanger 25 where third fluid is circulated in heat exchange relationship with said first fluid. Then said first fluid is further accelerated and compressed with vanes 30 assuring that said first fluid will rotate with said rotor. Heat is added to said first fluid in heat exchanger 12, where said second fluid is circulated in heat exchange relationship with said first fluid. After compression, said first fluid is passed through nozzles 13 where said first fluid is further accelerated and discharged in forward direction so that the absolute first fluid tangential velocity is the sum of the rotor velocity and the leaving fluid velocity; said first fluid is

passed to space 24, and from there to inward extending first fluid passageways where vanes 15 will assure that said first fluid will rotate with said rotor for receiving the work associated with deceleration of said first fluid by said rotor. After deceleration, said first fluid is passed through passage 18 thus completing its work cycle.

In the turbine of this invention heat is removed during or before the first part of compression, and heat is added into said first fluid during latter part of said compression. The removal of heat during compression may be sufficient to provide for isothermal compression during said heat removal, and this will allow for a greater work output by the turbine while having a lower temperature second fluid used for said heat addition. Thus, more work is obtained with a lesser temperature differential between said second fluid and said third fluid. Therefore, this turbine is particularly advantageous when the temperature of said second fluid is low, and near the temperature of said third fluid.

Various controls and governors are employed with the turbine of this invention. They do not form a part of this invention and are not further described herein.

The heat exchangers for said second and third fluids are shown to be made of finned tubing in the drawings. Other types of heat exchangers may be used, such as second fluid and third fluid conduits built in the rotor walls, for example.

The first fluid is usually a gas, such as many of the hydrocarbons. The second fluid may be water, or some other liquid or a gas, and the third fluid also may be a liquid, or a gas.

The heat removal heat exchanger 25 may be extended to the inward extending first fluid passages defined by vanes 15, if desired, to provide additional heat transfer area.

What is claimed is:

1. A turbine for generating power and comprising:
 - a. a means for supporting shaft rotatably;
 - b. a shaft journaled in bearings in said support for rotation;
 - c. a rotating rotor mounted on said shaft so as to rotate in unison therewith, said rotor being adapted for high speed rotation, said rotor having first radially outwardly extending first fluid passageways with vanes therewithin for ensuring that said first fluid therewithin rotates at the same rotational speed as said rotor for effecting centrifugal compression and for effecting an elevated pressure; said first radially extending passageways having at their outward ends means for discharging said first fluid in forward direction which is in the direction of rotation; said first radially extending passageways being provided with heating heat exchanger near the outward ends of said vanes to add heat to said first fluid prior of its said discharge forward; said first radially extending passageways being provided with cooling heat exchanger to remove heat from said first fluid near the inward ends of said vanes for removing heat from said first fluid before and during early part of said compression; said first fluid being discharged from said forward discharge means to radially inward extending first fluid passages, with said inward extending passages having vanes therewithin for receiving the work associated with deceleration of said first fluid; said inward extending first fluid passages outward ends being situ-

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ated outwardly from said first fluid forward discharge means; said first fluid is then passed through passages to said first outward extending first fluid passageways; said heating heat exchanger being provided with with a second fluid with said second fluid entering and leaving said rotor via passageways near the center of said rotor; said cooling heat exchanger being provided with a third fluid with said third fluid entering and leaving said rotor via passageways near the center of said rotor;

d. a first fluid being circulated within said rotor with a predetermined amount of said first fluid having been sealed within said rotor;

- e. a second fluid being circulated within said heating heat exchanger in heat exchange relationship with said first fluid;
- f. a third fluid being circulated within said cooling heat exchanger and being in heat exchange relationship with said first fluid.
2. The turbine of claim 1 wherein said first fluid forward discharge means are converging type nozzles.
3. The turbine of claim 1 wherein said heating fluid is a liquid.
4. The turbine of claim 1 wherein said heating fluid is a gas.

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[54] **TURBINE**

[76] Inventor: **Michael Eskeli**, 6220 Orchid Ln.,
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[22] Filed: **Aug. 7, 1973**

Primary Examiner—C. J. Husar

[21] Appl. No.: **386,273**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 185,060, Sept. 30,
1971, Pat. No. 3,758,233.

[52] U.S. Cl. **415/80; 415/1**

[51] Int. Cl. **F01d 1/18**

[58] Field of Search **415/80, 1**

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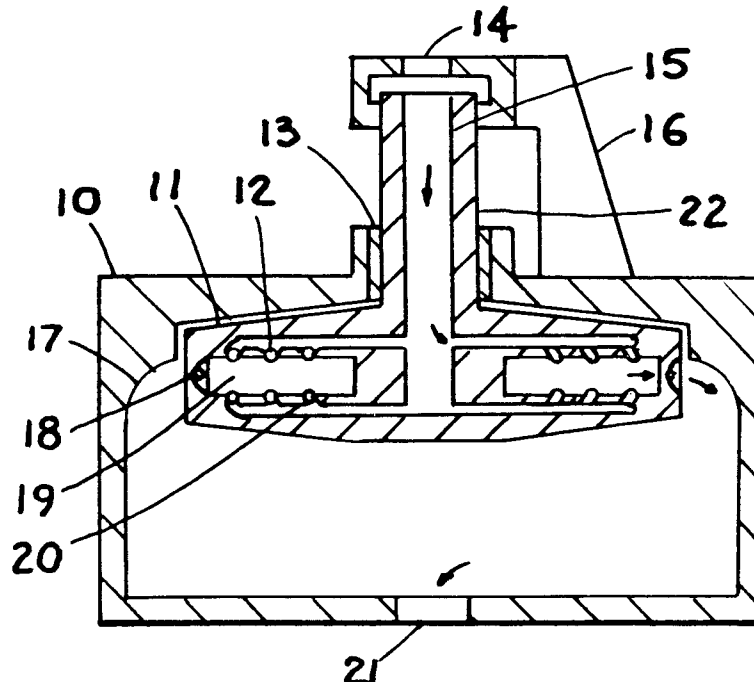
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[57] **ABSTRACT**

A method and apparatus for generation of power in response of a fluid flowing from a higher pressure to a lower pressure in rotating reaction turbine. Said fluid is supplied to said turbine rotor via nozzles in tangential direction forward at a velocity that is normally higher than the local turbine rotor velocity, and then said fluid discharged from said rotor near rotor periphery in backward direction thus generating thrust on said rotor which is then passed through said rotor shaft as the useful power output of the turbine. Fluids used may be gaseous, or liquids, including air and water.

10 Claims, 4 Drawing Figures



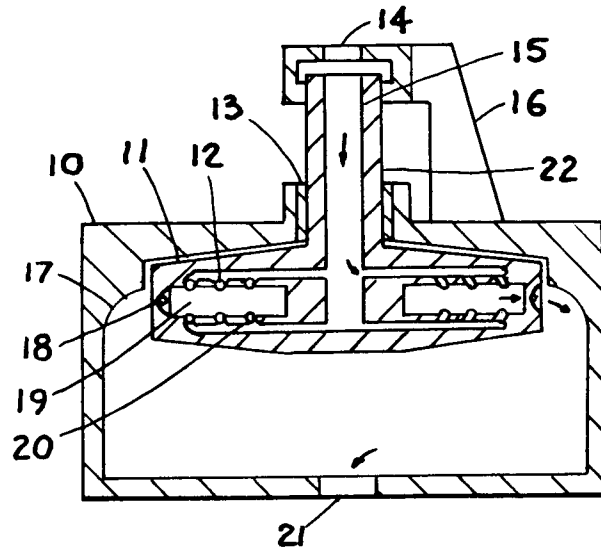


FIG. 1

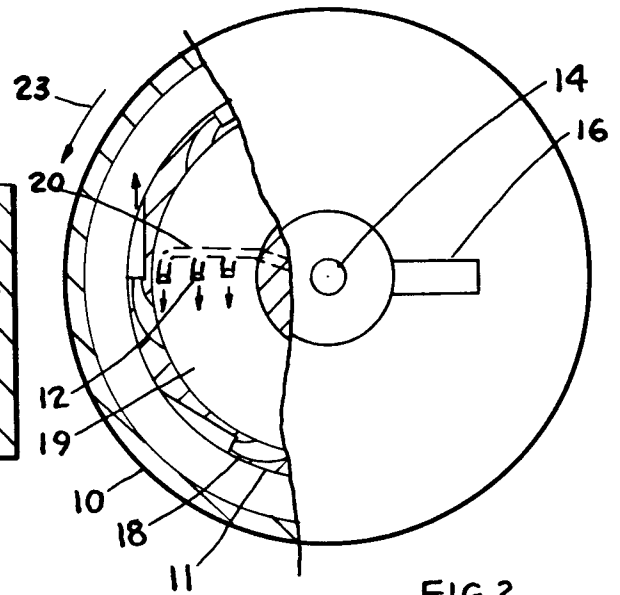


FIG. 2

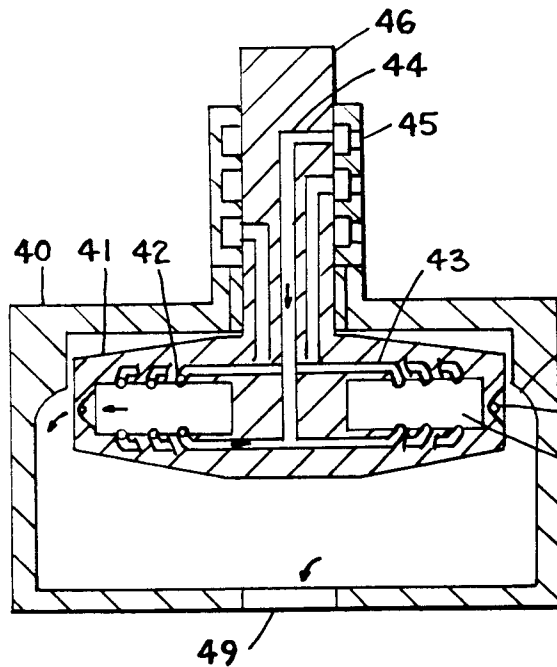


FIG. 3

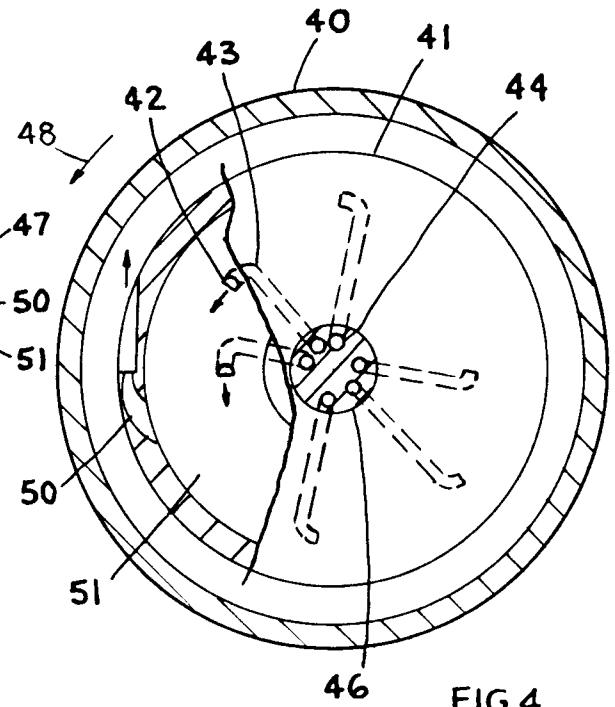


FIG. 4

TURBINE

CROSS REFERENCES TO RELATED APPLICATIONS

This is a continuing-in-part application of patent application titled "Reaction Rotor Turbine," filed Sept. 30, 1971, Ser. No. 185,060, by Michael Eskeli now U.S. Pat. No. 3,758,233.

The principles used with the turbine of this invention were also used in applications "Power Generator with Reaction Rotor," filed Feb. 2, 1973, Ser. No. 330,023; "Turbine," filed Feb. 8, 1973, Ser. No. 330,666; "Pressure Multiplier," filed Mar. 6, 1973, Ser. No. 338,436.

BACKGROUND OF THE INVENTION

This invention relates to power generation devices, and more particularly to turbines employing fluid reaction on the turbine rotor wheel as the power generating means.

The turbine of this invention differs from my earlier invention "Reaction Rotor Turbine," now U.S. Pat. No. 3,758,223, referenced hereinbefore, mainly in the manner that the fluid is introduced to the reaction rotor cavity. To better maintain control of the fluid velocity within the rotor cavity, the fluid is introduced to the rotor cavity via nozzles in the cavity walls where the fluid velocity can be controlled for each exit nozzle if so desired.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the device, and

FIG. 2 is an end view of the same unit with a section removed to show interior details.

In FIG. 3, a cross section of another form of the device is shown, and FIG. 4 is a section of FIG. 3, taken to illustrate the rotor and its interior.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide an improved means for generating power by employing a reaction rotor in a turbine, with a tangential fluid feed to the rotor cavity to compel the rotation of said fluid within said rotor cavity preferably at a higher absolute tangential velocity than the local absolute tangential velocity of the rotor; this will then result in an increased fluid pressure near the rotor periphery and thus in increased exit velocity for the fluid from said rotor with improved power output.

Referring to FIG. 1, therein is illustrated one form of the turbine, showing interior details of the unit. 10 is casing, 11 is rotor, 12 are fluid feed nozzles into rotor 11 cavity, 13 is rotor shaft bearing, 14 is fluid inlet to rotor shaft fluid passage 15, 16 is support, 17 is fluid deflector, 18 is fluid exit nozzle from rotor, 19 is rotor fluid cavity, 20 is fluid passage built to rotor wall, and communicating with rotor shaft passage 15 and fluid entry 14, 21 is fluid exit from casing 10, and 22 is rotor shaft.

In FIG. 2, another view of the unit shown in FIG. 1, is illustrated. 10 is casing, 11 is rotor, 18 is rotor fluid exit nozzle, 19 is rotor fluid cavity, 12 is fluid supply nozzle, 20 is fluid passage providing fluid to nozzle 12, 23 is indicator showing direction of rotation of the rotor, 14 is fluid inlet, 16 is support.

In FIG. 3, another form of the turbine is shown, where fluid supply nozzles to rotor cavity are supplied by separate fluid conduits thus allowing close control

of the fluid exit velocity from said nozzles into said rotor cavity. 40 is casing, 41 is rotor, 42 is fluid entry nozzle to rotor cavity, 43 is supply duct to nozzle 42, 44 is fluid passage in rotor shaft 46 communicating with supply passage 43, 45 is fluid entry, 47 is fluid deflector, 49 is fluid exit opening, 50 is fluid exit nozzle from rotor 41, 51 is rotor fluid cavity.

In FIG. 4, another view of the unit shown in FIG. 3, is illustrated. 40 is casing, 41 is rotor, 42 is fluid entry nozzle to rotor cavity 51, supplied by passage 43, 44 is shaft fluid passage, 46 is rotor shaft, 50 is rotor fluid exit nozzle, 48 is direction arrow indicating direction of rotation for rotor 40.

In operation, fluid is supplied to rotor through fluid passages located via rotor shaft and rotor walls, and through nozzles located in rotor walls discharging forward to the direction of rotation. Said fluid is pressurized within said rotor cavity due to it being forced to travel along a curved path, with higher pressure occurring near rotor periphery. This high pressure fluid is then discharged from the rotor through exit nozzles located near the rotor periphery; said discharge being backward away from the direction of rotation, thus said fluid will produce thrust on said rotor and produce torque on said rotor. This torque is then passed to said rotor shaft and to an external load as the useful power output of the turbine.

The absolute tangential velocity of the fluid entering the said rotor cavity is normally as high or higher than the absolute tangential velocity of the rotor in the area where the said fluid enters said rotor cavity. Normally, the tangential velocity of the fluid and the rotor are the same in the area near the rotor exit nozzles, such as nozzles 50 in FIG. 4; however, said velocity may also be different, as desired. In FIG. 2, the nozzles 12 are all supplied from a single fluid source; to effect differences in the fluid exit velocity from entry nozzles 12, suitable flow control devices may be provided in said nozzles or in their connecting fluid passages. Alternately, the velocity from all of the nozzles 12, may be allowed to vary; this variation is due to the partial pressurization of the fluid by centrifugal force due to rotation of the rotor. In FIG. 3 and FIG. 4, these entry nozzles 42 are fed separately from outside, and the exit velocities for the fluid from said entry nozzles 42 can be thus closely controlled by varying the fluid supply pressure. Usually, all the nozzles the same radial distance away from center of rotation, for the unit of FIG. 3, are provided with same pressure fluid; as shown in FIG. 3, three sets of nozzles are used and three separate fluid sources are used, each at its own pressure. Obviously, more or less sets of nozzles may be used, and also, each nozzle may be provided with a separately controlled fluid supply, if so desired.

Normally, the fluid absolute tangential velocity is greater than the rotor tangential velocity except at the periphery where they are the same. For example, for a fluid that is water, the rotor tangential velocity at periphery may be 100 feet per second, and the fluid tangential velocity at periphery also be 100 feet per second. However, the fluid tangential velocity may be kept at 100 feet per second constant at all points of the rotor cavity by controlled discharge velocity from said fluid entry nozzles; while, as in normal, the rotor wall tangential velocity decreases linearly to zero at the center of rotation. Thus, the fluid pressure at the periphery will be greater than is normally obtained in centrifuges,

and this will result in a higher exit velocity for the fluid from the rotor exit nozzles, and in more power output from the unit, when expressed per pound of fluid flowing.

The fluids used with this turbine may be either liquids or gases. Water or air may be used.

The rotor speeds for liquids are normally lower, while for gases, speeds up to the practical limits imposed by strength of various materials may be used. The rotors in the drawings are shown to have heavy material thicknesses near rotor center; this is normally required in rotors intended for high speed rotation. Also, the rotor is shaped like a disc so as to withstand high rotational speeds; other shapes may be used with units where lower operational speeds are employed.

Applications for this turbine include as a power generator in hydraulic dams, also, it can be used as a steam turbine. Further, it can be used in portable applications, such as a power generator for vehicles or for ships. Further, the fluid supply to the turbine rotor may be from a pump, or a compressor.

Various controls, governors and the like may be used with the turbine of this invention. They do not form a part of this invention and are not further described herein.

The fluid exit nozzles from said rotor are either converging or converging-diverging in shape as required for the fluid to obtain highest attainable fluid exit velocity from said rotor. Usually these fluid exit nozzles are sized and shaped as required to obtain isentropic expansion for said fluid. Fluid inlet nozzles to said rotor may be converging at their entry point to said rotor cavity, or they may be converging-diverging in shape. Usually, these fluid entry nozzles are not intended to accelerate the fluid except in a minor way for flow control purposes; the velocity of the fluid leaving these entry nozzles normally is approximately the same as the fluid velocity in adjoining fluid conduit furnishing said entry nozzles. However, acceleration in said entry nozzles may be provided for some fluids, especially for gaseous fluids.

What is claimed is:

1. A turbine stage for power generation responsive to flow of a fluid therethrough comprising:

a. a reaction rotor for producing said power; said rotor being rotated at a predetermined speed; said rotor having a plurality of reaction discharge passageways disposed near the periphery of said rotor and being oriented to discharge said fluid and generate torque on said rotor responsive to flow of fluid through said discharge passageways; said rotor being hollow and having a first cross sectional area at any radial distance radially exterior of a first fluid feeder nozzle that is greater than a second cross sectional area of said discharge passageways so as to effect a peripheral pressure that is greater than the pressure at the inlet to the rotor immediately downstream of a set of fluid feeder nozzles because said fluid is compressed by passing said fluid along a curved path within a rotating hollow cavity; said rotor having heavy duty construction sufficient to withstand high rotational speeds; said rotor also including a feeding means for feeding said fluid to said rotor cavity; said feeding means being a part of said rotor and rotating with said rotor; said feeding means including

- i. an inlet port disposed adjacent the center of said compressing centrifuge rotor for entrance of the fluid thereinto;
- ii. at least one entry passageway connected with said inlet port for conducting said fluid interiorly of said rotor;
- iii. a plurality of fluid feeder nozzles connected with said at least one entry passageway; said fluid feeder nozzles being physically connected with said rotor so as to rotate therewith and impart rotational speed to the entering said fluid, in addition to the flow velocity imparted by differential pressure; said fluid feeder nozzles being oriented and arranged to discharge said fluid forwardly in the direction of rotation of said rotating rotor and being supplied with said fluid at an elevated pressure via said inlet port;
- b. a casing support for providing support for said reaction rotor;
- c. a shaft journaled in said casing for supporting said reaction rotor; said shaft being connected with said rotor and rotating in unison therewith;
- d. bearing means disposed about said shaft for reducing friction; and
- e. fluid being flowed through said turbine stage from an inlet pressure upstream of said fluid feeder nozzles to discharge pressure downstream of said reaction discharge passageways; said inlet pressure being greater than said discharge pressure; said fluid being flowed from said feeder nozzles at a velocity greater in absolute value than the tangential velocity of said rotor at the same radial distance for effecting a greater power output on said shaft.

2. The turbine stage of claim 1 wherein said fluid feeder nozzles and said fluid pressure are adapted such that the exit velocity of the fluid from said feeder nozzles added to the tangential velocity of the rotor at the same radial spacing thereof has a constant absolute value within said rotor cavity; absolute value being velocity relative to the earth; constant velocity meaning that the combined fluid and rotor velocities are constant such that the absolute velocity of the fluid with respect to the earth is constant across said rotor cavity.

3. The turbine stage of claim 1 wherein the several feeder nozzles have separately pressure controlled fluid entry conduits communicating with outside fluid supply sources.

4. The turbine stage of claim 1 wherein a single fluid supply source is employed for all entry nozzles and wherein each nozzle has means for controlling fluid exit velocity to a predetermined value, relative to rotor.

5. The turbine stage of claim 1 wherein said fluid is a liquid.

6. The turbine stage of claim 1 wherein said fluid is a gas.

7. A method of generating power in a reaction turbine comprising:

- a. passing a fluid to a rotating rotor cavity via several fluid entry nozzles at sufficient initial velocity from each nozzle to cause said fluid to rotate within said rotating rotor cavity at an absolute velocity that is greater than said absolute rotor velocity at a predetermined distance from the center of rotation;
- b. causing said fluid to flow in arcuate radially outward path with sufficient angular velocity to cause sufficient centrifugal force to pressurize said fluid within said rotor such that the pressure of said fluid

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at the periphery of said rotor is greater than at the inlet to said rotor shaft entry; and discharging said fluid through reaction nozzles mounted at the periphery of the rotating rotor in a direction that is opposite to the tangential direction of rotation for producing thrust and generating torque on the rotor and its shaft for producing power.

8. The turbine stage of claim 1 wherein said casing sealingly encompasses said rotor and has a discharge aperture for flow therethrough of the fluid discharged from said rotor.

9. The turbine stage of claim 1 wherein said dis-

charge passageways comprise nozzles that are arranged and oriented to discharge said fluid backward in a direction opposite the direction of rotation of said rotor.

10. The turbine stage of claim 1 wherein said plurality of fluid feeder nozzles are disposed at a plurality of staged radial distances from the center of said rotor in order to allow reducing the velocity of said fluid exiting from the fluid feeder nozzle closest to said center of said rotor and still maintain said fluid at a velocity greater than said rotor at radial distances out to said discharge passageways.

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Eskeli calls this the basic heat pump patent

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United States Patent [19]

[11] 3,926,010

Eskeli

[45] Dec. 16, 1975

[54] ROTARY HEAT EXCHANGER

2,522,781 9/1950 Exner..... 62/499
2,529,765 9/1950 Exner..... 62/499

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee,
Houston, Tex. 77042

[22] Filed: **Jan. 24, 1974**

[21] Appl. No.: **436,124**

Primary Examiner—Albert W. Davis, Jr.
Assistant Examiner—Sheldon Richter
Attorney, Agent, or Firm—Jennings B. Thompson

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 393,571, Aug. 31, 1973.

[52] U.S. Cl. **62/401**; 62/499; 122/26;
126/247; 165/86; 165/88; 415/64; 415/114;
415/178; 416/96

[51] Int. Cl.² **F25B 3/00**

[58] Field of Search 62/86, 87, 401, 402, 499;
122/26; 165/86, 88; 126/247; 415/1, 64, 114,
177, 178, 179, 199 A; 416/95, 96

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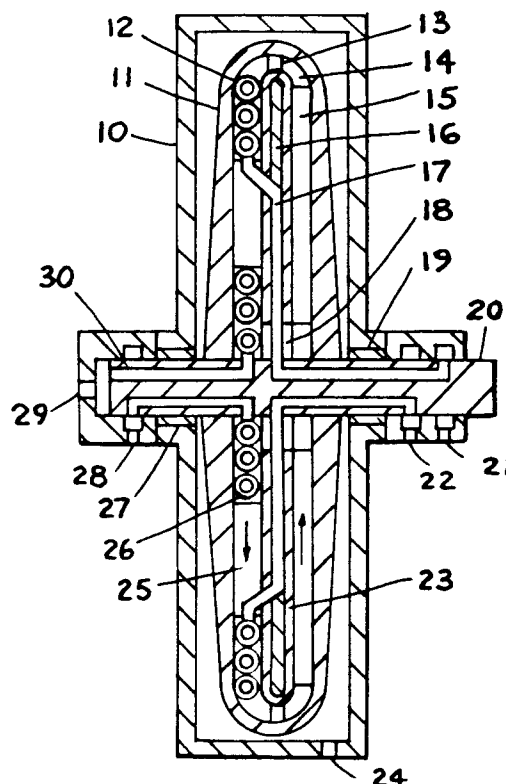
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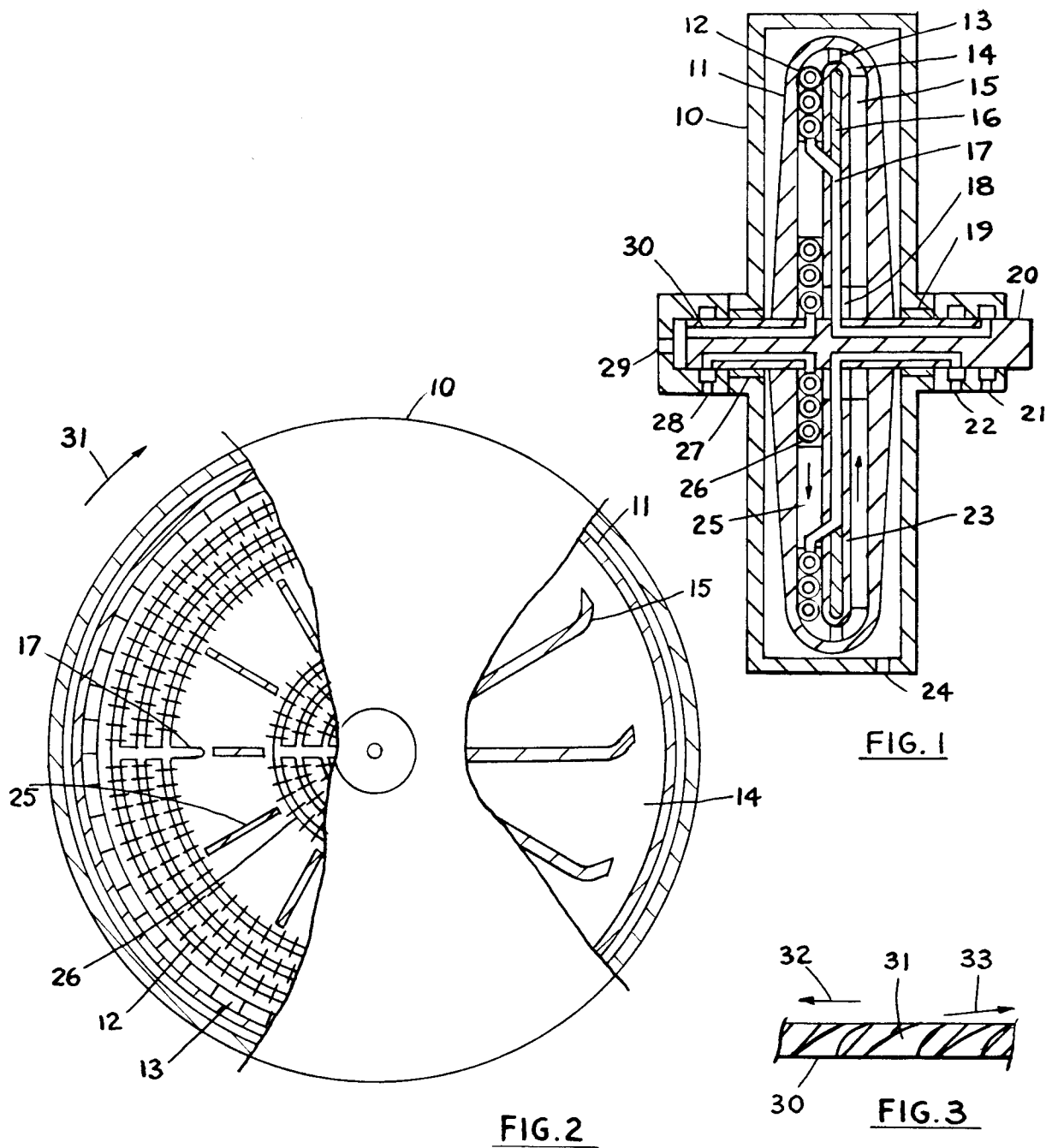
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[57] ABSTRACT

A method and apparatus for transferring heat from a fluid stream at a lower temperature to a fluid stream at a higher temperature. A gaseous fluid is sealed within a rotating rotor and circulated therein, compressing said gaseous fluid by centrifugal action on said fluid by said rotor, with heat being added to said gaseous fluid from said lower temperature fluid stream during first part of said compression, and then removing heat from said gaseous fluid during latter part of compression and after compression; said gaseous fluid having gained in temperature during said compression. Various fluids may be used for said lower temperature and higher temperature fluids, such as water, and said gaseous fluid may be carbon dioxide, or some other gas.

5 Claims, 4 Drawing Figures





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ROTARY HEAT EXCHANGER

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in part application of a previous application titled "Rotary Heat Exchanger," filed Aug. 31, 1973, Ser. No. 393,571. Also, the principles used with the heat exchanger of this invention were used previously with "Heat Exchanger with Three Fluids," filed May 17, 1973, Ser. No. 361,281, and "Heating and Cooling Wheel," filed Jan. 11, 1972, Ser. No. 216,938, and also "Heating and Cooling Wheel with Dual Rotors," filed Jan. 20, 1972, Ser. No. 219,212.

BACKGROUND OF THE INVENTION

This invention relates to devices for transferring heat from a fluid at a lower temperature to another fluid at a higher temperature by employing a compressible fluid which is compressed within a continuous flow centrifuge to an elevated pressure with accompanying temperature increase, and this higher temperature is then used to effect heat transfer to a second fluid which is at a higher temperature than said lower temperature fluid, and providing means within said centrifuge to propel said gaseous fluid through said centrifuge.

There have been several devices that have provided means of transferring heat from lower temperature fluid to a higher temperature fluid. These devices have been relatively inefficient due to the device requiring an external compressor to provide needed pressure differential to transport said gaseous compressible fluid through said centrifuge rotor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the heat exchanger, and

FIG. 2 is an end view of the unit shown in FIG. 1, with sections removed to illustrate interior details.

FIG. 3 is a detail showing fluid nozzles within the heat exchanger.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide a method and apparatus for transferring heat from a compressible and gaseous first fluid initially at a lower temperature to a second fluid at a higher temperature by compressing said first fluid within a centrifuge with accompanying temperature increase so that said first fluid is at a higher temperature when compressed than said second fluid thus providing needed temperature differential; said first fluid then being allowed to expand within said centrifuge with accompanying temperature decrease; said first fluid being passed via a set of nozzles arranged to discharge backward near the periphery of said centrifuge rotor; said gaseous first fluid having preferably heat added to it during and before compression, and having heat removed from said first fluid during and after compression.

Referring to FIG. 1, therein is shown a cross section of the heat exchanger. In this unit, said first fluid is sealed within the rotor, and the second fluid and the third fluid are supplied from external sources through passages provided within the rotor shaft, and then distributed to their respective heat exchangers. 10 is casing, 11 is rotor, 12 is heat removal heat exchanger

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through which said second fluid is circulated, 13 are first fluid nozzles, 14 is a space for said first fluid downstream of said nozzles 13, 15 are vanes within inward extending first fluid passages, 16 is thermal insulation, 17 is distribution conduit for second fluid, 18 is first fluid passage near rotor center for passing first fluid from expansion side to compression side, 19 and 27 are shaft bearings and seals, 20 is rotor shaft, 21 and 22 are second fluid entry and exit, 23 is rotor divider, 24 is casing vent into which a vacuum pump may be connected, 25 are compression side vanes, 26 is heat addition heat exchanger, wherein said third fluid is being circulated, 28 and 29 are third fluid entry and exit, 30 is third fluid passage within rotor shaft.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated. 10 is casing, 11 is rotor, 15 is vane, 14 is first fluid space, 13 are first fluid nozzles, 12 is heat removal heat exchanger, 25 are vanes, 26 is heat addition heat exchanger, 17 is second fluid conduit, 31 indicates direction of rotation for rotor.

In FIG. 3, a detail of first fluid rotor nozzles is shown. 30 is nozzle wall, 32 indicates direction of movement of nozzles, 13 are nozzles, and 33 indicates first fluid leaving said nozzles 13.

The operation of the heat exchanger is as follows: First fluid enters the compression side of the rotor via entry opening 18, and is compressed by rotor centrifugal action with accompanying temperature increase, with vanes 25 and fins of heat exchanger tubes assuring that said first fluid will rotate with said rotor. During first part of said compression, heat is added to said first fluid in heat addition heat exchanger 26, and then said first fluid is further compressed and then heat is removed from said first fluid and transferred to said second fluid in heat exchanger 12. After said heat removal, said first fluid is passed to expansion side of rotor via nozzles 13, with said nozzles discharging usually said first fluid backward away from direction of rotation thus providing additional pressure differential to provide for circulation of said first fluid within said rotating rotor. Said first fluid is then passed inward toward rotor center with vanes 15 assuring that said first fluid will rotate with said rotor for receiving the work associated with the deceleration of said first fluid. After said deceleration and expansion, said first fluid is passed via opening 18 thus completing its cycle. Heat is carried away from said heat exchanger by said second fluid through said rotor shaft passages, and heat is supplied to said first fluid by said third fluid being circulated via said rotor shaft passages. Shaft 20 is used to connect said rotor to a power source.

The unit described herein is similar to the unit described in co-pending patent application "Rotary Heat Exchanger," except that the placement of the heat addition heat exchanger is slightly different. The function of the heat exchanger is the same as in said co-pending application, which is Ser. No. 393,571.

The heat removal heat exchanger 12 is shown to be within the compression side of the said rotor; said heat exchanger may also be extended to be partially or fully within said expansion side of said rotor, without changing the function of said rotary heat exchanger. Similarly, said heat addition heat exchanger may be partially placed within the inward extending expansion side passages without changing the function of said rotary heat exchanger.

The rotor nozzles 13 may be arranged to discharge said first fluid radially or axially, if desired, depending

of the requirements of the first fluid used.

The said third fluid may be used for cooling applications, if desired; normally, when said third fluid is used for cooling, the said first fluid and said third fluid are arranged to be in counter-flow within said heat addition heat exchanger 26. Said second fluid may be vaporized within said heat removal heat exchanger, if desired.

Thermal insulation is provided as shown and as desired to prevent undesirable heat transfer between fluids and rotor.

The heat exchangers are shown having been made using finned tubing as the heat exchange members. Other forms of heat exchangers may be used if desired.

Various controls and governors may be used with the device of this invention. They do not form a part of this invention and are not further described herein.

Vanes 15 are shown to be curved in FIG. 2. Said vanes may be also made radial as desired. Further, vanes 25 may be curved if desired, and the fins of heat exchanger tubing slanted if desired.

The first fluid is usually a gas, such as carbon dioxide, or be a vapor, such as many of the halogenated hydrocarbons. The second fluid may be either gas or a liquid; normally said second fluid is a liquid. Said third fluid may be also either gas or a liquid. Water may be used as said second and said third fluid.

Work input to said rotor to rotate said rotor is normally low and may be negligible. Work is required to accelerate said first fluid to rotor tangential speed, and then work is recovered when said first fluid is decelerated. Discharge of said first fluid backward in nozzles 13 increases work loss, but if said nozzles are arranged to discharge said first fluid radially or in axial direction, this work loss can be eliminated. The amount of work lost in nozzles 13 depends on the fluid chosen for said first fluid, and of the location of said heat exchangers

12 and 26, and the temperature differences maintained within the rotor.

What is claimed is:

1. A heat exchanger comprising a rotor, means for mounting said rotor for rotation, said rotor having first and second closed passages extending outwardly from the axis of rotation of the rotor, a first passage means for connecting the outer ends and a second passage means for connecting the inner ends of said first and second closed outwardly extending passages to allow a fluid to flow outwardly in said first passage and inwardly toward the axis of rotation in said second passage, a compressible first fluid in said passages, a first heat exchanger carried by said rotor and a second heat exchanger carried by said rotor and located inwardly from the outermost part of said first heat exchanger, means for passing a second fluid through said first heat exchanger to remove heat from the first fluid, and means for passing a third fluid through said second heat exchanger to add heat to said first fluid, and means for rotating said rotor to cause said first fluid to flow outwardly in said first passage and be heated by compression due to centrifugal force and to flow inwardly in said second passage toward the axis of rotation of the rotor.

2. The heat exchanger of claim 1 in which the passage means connecting the outer ends of the passages comprises a set of nozzles for passing said first fluid from the first passage to the second passage.

3. The heat exchanger of claim 2 in which the nozzles direct the fluid flowing therethrough in a direction away from the direction of rotation of the rotor.

4. The heat exchanger of claim 2 in which the first heat exchanger is located upstream of the nozzles.

5. The heat exchanger of claim 4 in which the second heat exchanger is located in the same passage as the first heat exchanger.

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[54] COMPRESSING CENTRIFUGE

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee,
Houston, Tex. 77042

[22] Filed: Oct. 11, 1973

[21] Appl. No.: 405,584

[52] U.S. Cl. 62/401; 126/247; 165/86;
415/1; 415/64; 415/178; 416/96

[51] Int. Cl. F25b 3/00

[58] Field of Search 62/86, 87, 401, 402, 499;
122/26; 126/247; 165/86, 88; 415/1, 64, 114,
177, 178, 179, 199 A; 416/95, 96

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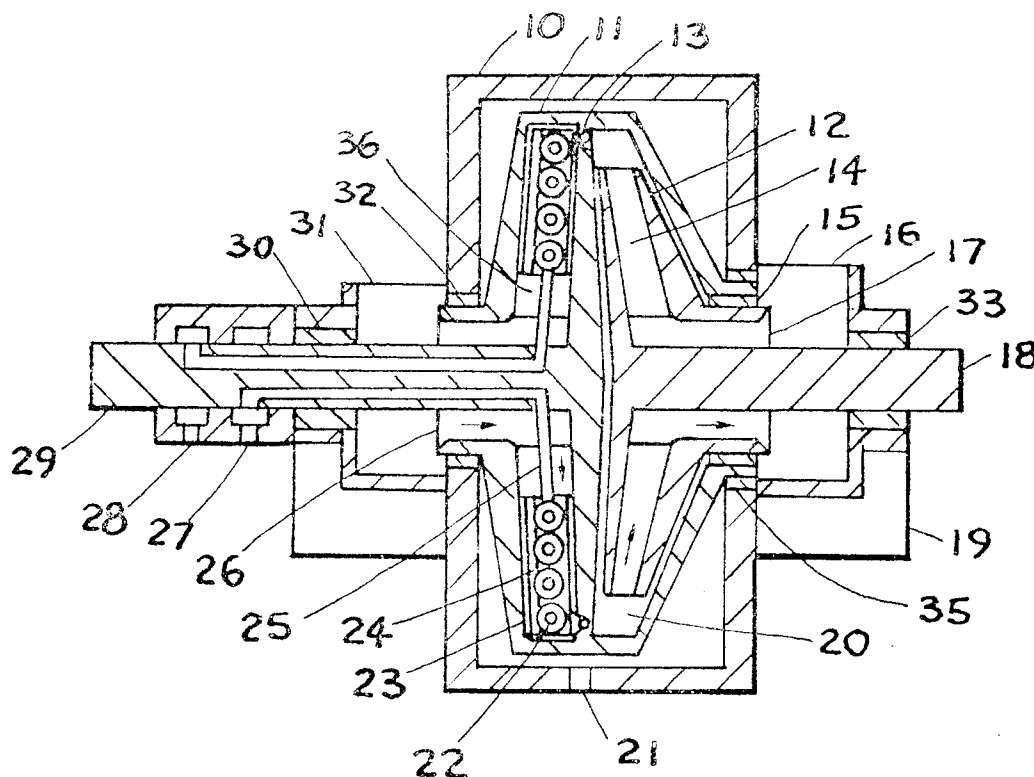
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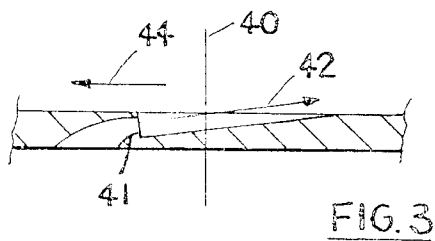
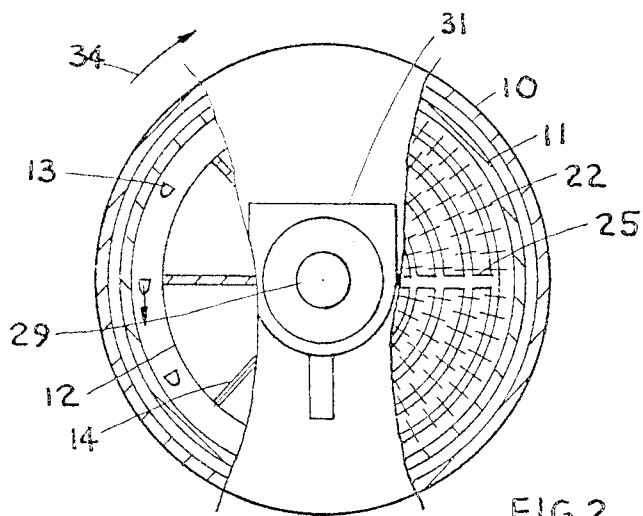
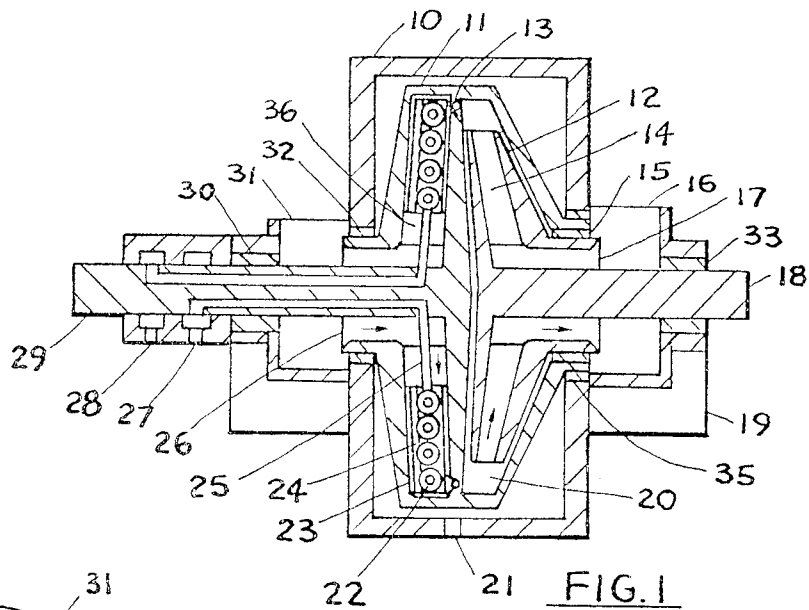
[57] ABSTRACT

A method and apparatus for compressing fluid by passing said fluid through a rotating continuous flow centrifuge wherein said fluid is pressurized by centrifugal action on said fluid by said centrifuge rotor. Said rotor is provided with passageways for said fluid with vanes placed therewithin for assuring that the fluid will rotate with said rotor. After compressing occurs, said fluid is passed in compressed state to a second rotor wherein energy contained in said fluid is converted to work, with said fluid being passed within fluid passageways in said second rotor inwardly to an exit at center, with vanes ensuring that said fluid second rotor will rotate with said fluid for receiving the work associated with deceleration of said fluid. Cooling may be provided during compression of said fluid by circulating a coolant in heat exchange relationship with said fluid. Nozzles may be provided near the first rotor periphery for decelerating said first fluid. Fluids being compressed may be gases or gases with minor amounts of liquid, such as air, or halogenated hydrocarbons.

Primary Examiner—Albert W. Davis, Jr.
Assistant Examiner—Sheldon Richter
Attorney, Agent, or Firm—Robert W. Dickerson

7 Claims, 3 Drawing Figures





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COMPRESSING CENTRIFUGE

CROSS REFERENCES TO RELATED APPLICATIONS

This application is similar to my U.S. Pat. No. 3,761,195, and 3,793,848 with some additions and improvements. (Compressing Centrifuge)

Also, similar principles were used in patent application "Compressing Centrifuge with Cooling", Ser. No. 243,239, filed 4/12/72.

BACKGROUND OF THE INVENTION

This invention relates to devices for compressing gases from a lower pressure to a higher pressure by employing a continuous flow centrifuge as the compressing first rotor and using an inward flow turbine as the second rotor wherein energy contained in the fluid leaving said first rotor is converted to work.

In my previous U.S. Pat. No. 3,761,195, I described a similar compressor to the compressor of this invention; the device of this invention adds some improvements and changes to said previously patented compressor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of the compressor, and

FIG. 2 is an end view of the same unit with portions removed to show internal details;

FIG. 3 is a detail of the nozzles employed within first rotor.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide for partial reduction in the absolute tangential velocity of the working fluid by providing nozzles oriented backward near the periphery of said a first rotor through which nozzles said working fluid will pass to a second rotor. Further, it is an object of this invention to provide a cooling heat exchanger within said first rotor to remove heat of compression either fully or in part. It is also an object of this invention to provide a casing around said rotors to allow evacuation of said casing thus reducing fluid friction on said rotor external surfaces.

Referring to FIG. 1, therein is shown a cross section of the compressor. 10 is casing, 11 is first rotor, 12 is second rotor, 13 are first rotor exit nozzles, 14 are second rotor vanes, 15 is rotor seal, 16 is working fluid exit from unit, 17 is working fluid exit from second rotor, 33 is second rotor shaft bearing, 18 is second rotor shaft, 19 is shaft support, 20 is space within first rotor, 21 is casing aperture into which a vacuum pump may be connected, 22 is cooling heat exchanger, 23 is a layer of thermal insulation, 24 is heat exchanger support plate, 25 is coolant distribution conduit, 26 is working fluid entry to first rotor, 27 and 28 are coolant entry and exit, 29 is first rotor shaft supported by bearing 30, 31 is working fluid entry to unit, 32 is rotor seal, 36 is vane within first rotor. 35 is a seal.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated with portions removed to show interior details. 10 is casing, 11 is first rotor, 22 is heat exchanger, 25 is coolant distribution conduit, 14 is second rotor vane, 12 is second rotor, 29 is shaft, 13 are nozzles, 34 indicates direction of rotation for both rotors. 31 is fluid entry.

In FIG. 3, a detail of the first rotor nozzles is shown. 40 is shaft direction about which rotor rotates, 44 is

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direction of rotation, 41 is nozzle, and 42 indicates leaving fluid.

In operation, the working fluid enters unit via opening 31 and is passed to first rotor via entry post 26, and then said working fluid passes to rotor interior where vanes 36 and heat exchanger 22 will assure that said working fluid will rotate with said rotor. After compression, said working fluid leaves said first rotor via nozzles 13 or more particularly as indicated in FIG. 3 by numerals 41, 42; these nozzles are oriented to discharge said working fluid backward, thus reducing the absolute tangential velocity of said fluid. Said fluid then enters said second rotor 12, where vanes 14 by virtue of the working fluid acting thereagainst will assure that said rotor will rotate with said working fluid; said working fluid is then discharged from said second rotor via exit port 17, and from the unit via opening 16.

Cooling may be applied to said working fluid during compression, by circulating a coolant through said heat exchanger 22, in heat exchange relationship with said working fluid. Coolant enters via entry 27 and thence through said first rotor shaft passages, and leaves via passages through said first rotor shaft to exit through exit 28.

Work is required to accelerate said working fluid to the tangential speed of said first rotor; power is supplied to the shaft of said first rotor. Some work is returned to said first rotor in nozzles 13, and additional work is produced by the decelerating working fluid in second rotor passages. Normally, the rotational speed of said second rotor is less than said first rotor. Also, in normal operation, the rotational speed of said first rotor is sufficient to provide for a substantial exit velocity for said working fluid from said rotor nozzles 13; then also, the rotational speed of said second rotor is substantially less than said rotational speed of said first rotor. The difference in speeds for the two rotors is normally fixed and the two rotors are connected by suitable external power transmission means to pass the work produced by said second rotor to said first rotor, with additional work for rotating said first rotor being provided from an external power source.

If the cooling heat exchanger is not provided, then the vanes 36 are extended to the periphery of said first rotor to assure that said working fluid will rotate with said first rotor.

The first rotor is shown in FIG. 1 to have been extended to inclose said second rotor, with seals 35 and 15 providing sealing to maintain working fluid pressure within rotor and to seal with casing. Space 20, FIG. 1, is provided to allow working fluid leaving nozzles 13 to move and adjust to second rotor tip speed. It should be noted that usually the absolute working fluid tangential velocity when entering said second rotor is the same as the tip speed of said second rotor, for best efficiency.

Applications of this compressor include use as an air compressor, as a compressor for various gases, and as a refrigeration compressor and heat pump compressor. Also, it can be used as the compression stage in power generation devices.

Various governors and controls are used with this compressor. They do not form a part of this invention and are not further described herein.

The rotors for the compressor of this invention are shaped to allow high rotational speeds, with rotor walls made of thick material sections and generally having thick center sections. The heat exchanger is made of finned tubing attached to a support plate to prevent

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movement and crushing; said support plate is then supported by said rotor walls. Other types of heat exchanger construction may be employed, such having coolant passages built to rotor walls. Also, the vanes 14 of said second rotor may be made with suitable curvature if said working fluid tangential velocity should differ from the second rotor tangential tip velocity.

The radial velocity of the working fluid within the radially extending passageways of both rotors is normally slow, so that the compression within said first rotor is nearly non-flow type, and the expansion within said second rotor is nearly non-flow type. Thus the compression within said first rotor is similar to compression within a centrifuge rotor.

What is claimed is:

1. A working fluid compressor expander comprising:

- a. a casing for supporting shafts;
- b. shafts journaled in bearings in said casing for rotation;

c. a rotating first rotor mounted on a first rotor shaft so as to rotate in unison therewith, said rotor being of circular cross section in a radial plane and adapted for high speed rotation with its structural walls being thicker near the center than at its periphery; said rotor having an entry port for said working fluid near the center of said rotor and radially extending passageways having vanes there-within for assuring that said working fluid there-within rotates at the same rotational speed as said rotor for effecting centrifugal compression and effecting an elevated pressure at its periphery a cooling heat exchanger within said radially extending passageways in said first rotor with a coolant being circulated through passages within said heat exchanger in heat exchange relationship with said working fluid for the purpose of removing heat from said working fluid; said coolant being supplied to said heat exchanger via passages provided through said first rotor shaft, and returned via passages within said first rotor shaft; nozzles communicating at their entry end with said radially extending passageways and discharging said working fluid to a peripheral portion within said rotor; with said

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peripheral portion being separated from said radially extending passage ways by a wall with said nozzles mounted on said wall;

- d. a rotating second rotor mounted on a second rotor shaft so as to rotate in unison therewith, said second rotor being of circular configuration in a radial plane and adapted for high speed rotation; said rotor having radially extending passageways communicating with said peripheral portion of said first rotor for receiving said working fluid; said radially extending passageways extending inwardly toward the center of said rotor, and having vanes there-within to cause rotation of said second rotor on impingement by said working fluid against said vanes; and having an exit port near the center of said second rotor.

2. The compressor of claim 1 wherein said nozzles mounted within said first rotor are oriented to discharge said working fluid tangentially backward in a direction away from the direction of rotation of said first rotor thus reducing the absolute tangential velocity of said working fluid.

3. The compressor of claim 1 wherein said casing is provided with an aperture for evacuating a space between said first rotor and said casing for reducing fluid friction on said first rotor.

4. The compressor of claim 1 wherein the tangential velocity of the second rotor outer periphery is less than the tangential velocity of the nozzles mounted on said first rotor.

5. The compressor of claim 1 wherein the tangential velocity of the outward ends of the second rotor vanes is the same as the tangential velocity of the working fluid entering said radially inward extending passageways of said second rotor.

6. The compressor of claim 1 wherein the tangential velocity of the outward ends of the second rotor vanes is less than the tangential velocity of the working fluid entering said radially inward extending passageways of said second rotor.

7. The compressor of claim 1 wherein said working fluid is a gas with minor amounts of liquid, if any.

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[54] MULTISTAGE HEAT EXCHANGER

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee,
Houston, Tex. 77042

[22] Filed: Jan. 2, 1974

[21] Appl. No.: 429,496

[52] U.S. Cl. 62/401; 165/88; 165/140;
415/1

[51] Int. Cl.² F25B 3/00

[58] **Field of Search** 165/86, 88, 140; 62/86,
62/87, 401, 402, 499; 415/1

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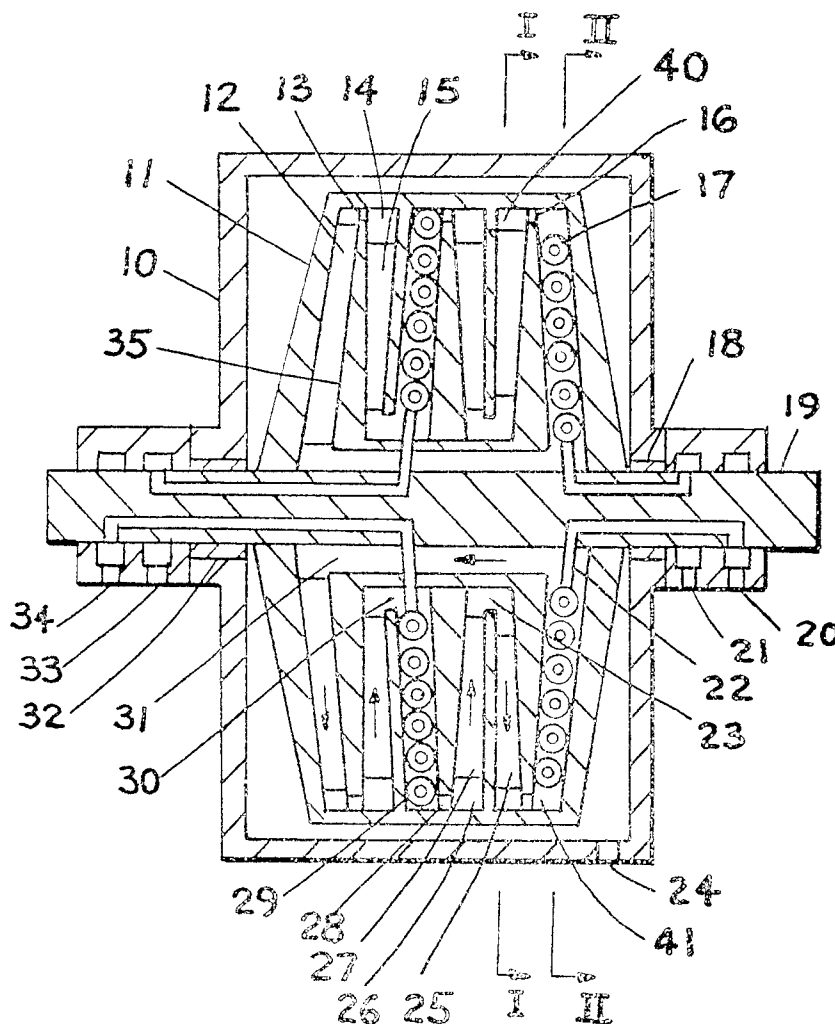
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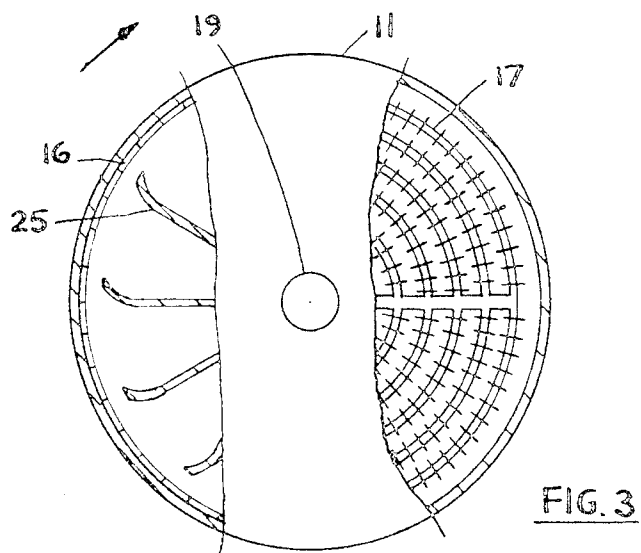
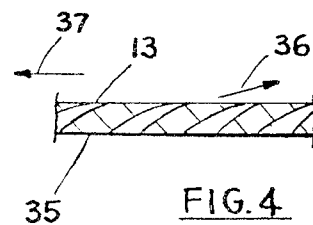
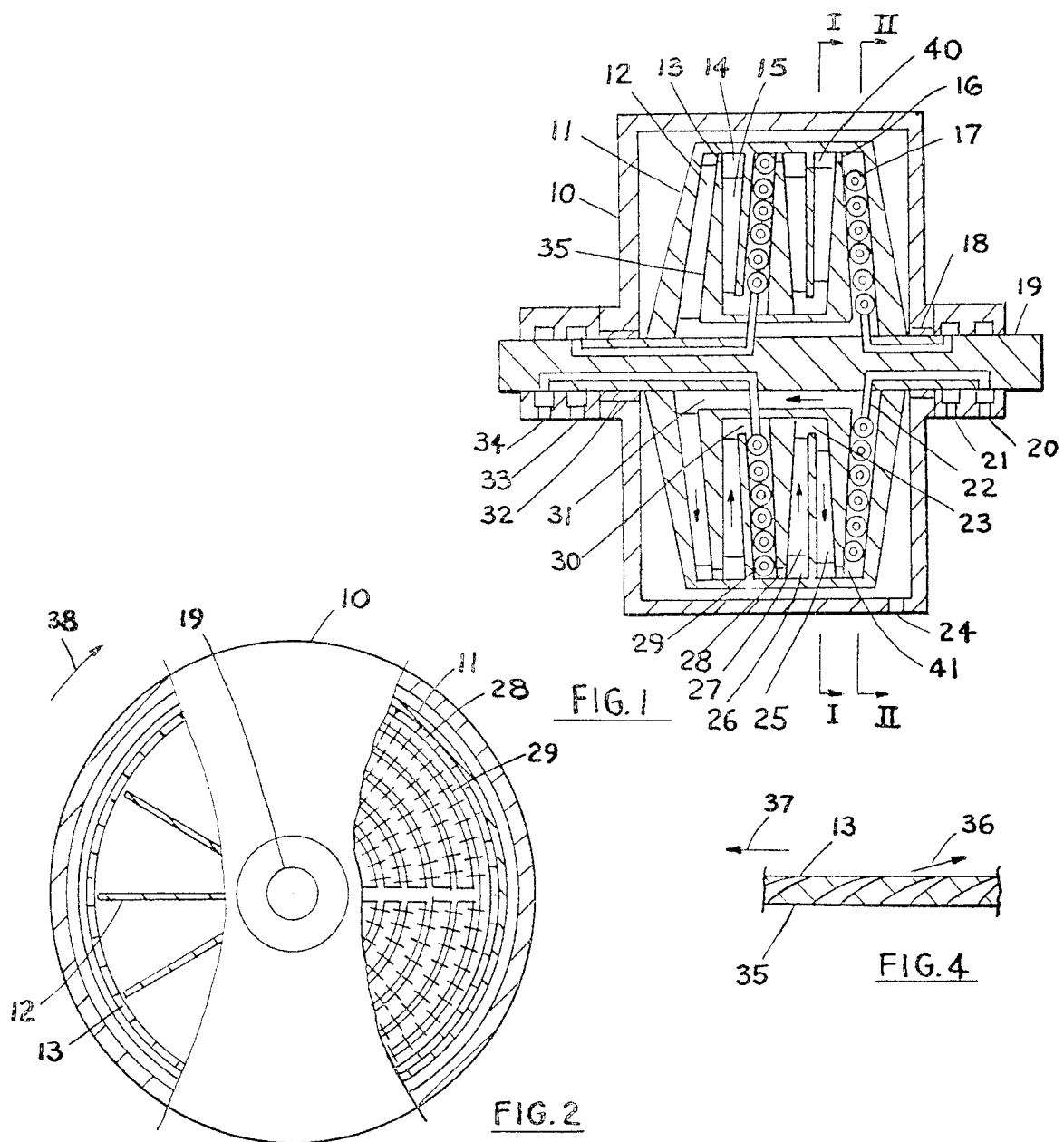
Primary Examiner—Albert W. Davis, Jr.
Assistant Examiner—Sheldon Richter

[57] ABSTRACT

A method and apparatus for transferring heat from a source at a lower temperature to a fluid at a higher temperature by employing a rotating centrifuge and a compressor to compress a gaseous first fluid with accompanying temperature increase and then transferring heat from said first fluid to a second fluid being in heat exchange relationship with said first fluid within said centrifuge. The first fluid is then allowed to expand within the rotor and heat is added during and after expansion from a third fluid being in heat exchange relationship with said first fluid. The first fluid is normally a gas, the second fluid may be a gas or a liquid, and the third fluid may be a gas or a liquid. Typical first fluid may be carbon dioxide, second fluid be water, and third fluid be water.

2 Claims, 4 Drawing Figures





MULTISTAGE HEAT EXCHANGER

CROSS REFERENCES TO RELATED APPLICATIONS

This application uses principles similar to those used in "Rotary Heat Exchanger" filed 8/31/73, Ser. No. 393,571, and "Heating and Cooling Wheel", filed 1/11/72, Ser. No. 216,938.

This invention relates generally to devices for transferring heat from a fluid at a lower temperature to another fluid at a higher temperature by employing a compressible fluid which is compressed within a continuous flow centrifuge to an elevated pressure with accompanying temperature increase, and this higher temperature is then used to effect heat transfer to a second fluid which is at a higher temperature than said lower temperature fluid, and providing means within said centrifuge to propel said gaseous compressible fluid through said centrifuge.

There have been several devices that have provided means for transferring heat from a lower temperature fluid to a higher temperature fluid. These devices have been inefficient due to the need to employ an external compressor to provide a pressure differential to transport the gaseous compressible first fluid through said centrifuge rotor; also, they have been incapable of producing the high temperatures needed to produce high pressure water steam due to inherent stress limitations imposed by materials of construction for the rotor.

FIG. 1 is a cross section of the heat exchanger, and FIG. 2 is an end view of the unit shown in FIG. 1;

FIG. 3 is an end view of the rotor, showing some interior details, and

FIG. 4 is a section of rotor internal nozzles.

It is an object of this invention to provide a means for producing high temperature fluids while using low temperature heat sources wherein the said first fluid is first compressed in a compression stage prior to passage to heat exchanger stage, and then expanding said first fluid in more than a single stage with addition of heat to said fluid during and after said expansion. The purpose of having multiple compression and expansion stages is to reduce the rotational speed of the rotor so that standard and readily available materials may be used for rotor construction, and also to extend the life of bearings and seals.

Referring to FIG. 1, therein is shown a cross section of the multi-stage rotary heat exchanger. 10 is casing, 11 is rotor, 12 is compressor stage vane, 13 are first nozzles, 14 is first fluid space, 15 are second vanes, 16 are third nozzles, 17 is heat addition heat exchanger, 18 is bearing and seal, 19 is rotor shaft, 20 and 21 are third fluid inlet and outlet, 22 is third fluid conduit, 23 is first fluid passage opening, 24 is casing vent into which a vacuum source may be attached, 25 are fourth vanes, 26 is second fluid space, 27 is third fluid space, 28 is a fluid space, 29 are third vanes, 30 are second first fluid nozzles, 31 is heat removal heat exchanger, 32 is first fluid passage, 33 is first fluid return passage, 34 is rotor shaft bearing and seal, 35 and 36 are second fluid entry and exit, 37 is a rotor interior dividing wall. Sections I and II are further illustrated in FIG. 3.

In FIG. 2, an end view of the unit is shown with portions removed to show rotor interior details. 10 is casing, 11 is rotor, 28 are third nozzles, 29 is heat removal

heat exchanger, 13 are first nozzles, 12 are first vanes, 38 indicates direction of rotation, and 19 is shaft.

In FIG. 3, and end view of the rotor is shown with additional interior details. 11 is rotor, 17 is heat addition heat exchanger, 25 are fourth vanes, 16 are third nozzles and 19 is rotor shaft.

In FIG. 4, a detail of rotor nozzles is shown. 37 indicates direction of rotation when the nozzles are used in compressing section such as nozzles 13 and 28 in FIG. 1, and 36 indicates fluid leaving said nozzles. 35 is rotor divider wall, and 13 is nozzle. The third nozzles 16 are similar but oriented in opposite direction so that the first fluid leaves said nozzles in forward direction that is in the direction of rotation.

In operation, first fluid is first compressed in rotor outward extending passages defined by first vanes 12, and then passed via first nozzles 13 in backward direction that is away from direction of rotation so that said first fluid will have an absolute tangential velocity that is less than said rotor tangential velocity in same area. After leaving said first nozzles 13, first fluid is decelerated further in second vanes 15, and then passes via passage 30 near rotor center to be further compressed by passages formed by heat exchanger 29 fins which act as vanes, and at the same time heat is removed from said first fluid in a predetermined amount into a second fluid being circulated within said heat exchanger 29. After passing through the passage of heat exchanger 29, said first fluid is passed through second nozzles 28 that are oriented to discharge said first fluid in a backward direction and then passing to inward passages formed by vanes 27 and then passing through passage 23 near rotor center, after this said first fluid is accelerated in outward passages formed by vanes 25 and then said first fluid is decelerated in inward rotor passages formed by heat addition heat exchanger 17 fins; after this said first fluid is returned to be again compressed via passage 31 thus completing the work cycle for said first fluid. Said second fluid is supplied from external sources and passed through passages within rotor shaft to said heat exchanger 29 and then said second fluid is returned via said rotor shaft to exit. The said third fluid is also circulated via said rotor shaft to the heat addition heat exchanger 17 and then returned via said rotor shaft to exit. The temperature of leaving second fluid is greater than the temperature of entering second fluid. The temperature of said third fluid entering is greater than the temperature of leaving third fluid.

The heat exchanger of this invention uses two stages of compression and two stages of expansion; additional compression and expansion stages may be used if desired to even further reduce the rotor rotational speed and to increase leaving second fluid temperature.

For some fluids, being used as said first fluid, nozzles 28 may be omitted, and plain openings substituted. Alternately, the second nozzles 28 may be arranged to discharge said first fluid forward so that the tangential velocity of leaving first fluid is greater than the tangential velocity of said rotor in the area. Also, the location of inner ends and outer ends of said first vanes 12, said second vanes 15, said third vanes 27 and said fourth vanes 25 may be as desired to obtain required pressure conditions within the rotor so that said first fluid is transported through said rotor passages and pressurized to the desired pressure, with a minimum work input to said rotor shaft from external sources. Further, it should be noted that heat exchangers 17 and 29 may have lesser amount of finned coils so that part of the

respective first fluid passage is left open; in these instances, vanes are normally added within said first fluid passages to assure that said first fluid will rotate with said rotor. Also, the heat exchanger 29 may be extended into passages defined by vanes 12, if desired by design conditions for the first fluid being used. Similarly, heat exchanger 17 may be extended to passages defined by vanes 27 and 25 if desired by design conditions. Further, said heat exchanger 17 may be also extended to first fluid passage 31 if desired thus adding to heat exchanger area.

The first fluid used for the heat exchanger of this invention should be a gaseous and compressible fluid. Gases such as carbon dioxide, nitrogen or air may be used, and many of the halogenated hydrocarbons may also be used. Normally, the gas being used should not condense while being within the rotor, since condensation will block the first fluid passages. The second fluid may be a liquid or a gas; usually said second fluid is a liquid. The third fluid also may be either a liquid or a gas.

The heat exchanger coils within the rotor are shown having been made of finned tubing; other types of heat exchangers may be used, including types where said second fluid and said third fluid passages are built to rotor walls, or where metal heat conducting parts are employed to transfer heat from one fluid to another.

First fluid spaces 14, 26 and 41 are provided at discharge side of nozzles to provide a space for the first fluid to be discharged into.

Various governors and controls may be used with the heat exchanger of this invention. They do not form a part of this invention and are not further described.

Vanes 12, 15, 27 and 25 may be curved in conventional manner if desired. Similarly, the fins of the heat exchangers may be slanted if desired; for heat exchanger 29 they may be slanted backward if desired and for heat exchanger 17 slanted forward.

It should be noted that the first fluid is compressed within passageways defined by vanes 12, and also accelerated, and then the first fluid is decelerated in nozzles 13 and further decelerated in vanes 15. After passing to opening 30, the first fluid will be at a pressure that is greater than the pressure in passage 31, while the rotational speed of the first fluid has been reduced to a minor value existing in the area. The first fluid is then again accelerated in the second outward extending passageways wherein heat exchanger 29 is located, with a pressure increase. The first fluid is then passed through openings 28, which may be nozzles, or plain holes, depending on the design. Similarly, openings 16 may be nozzles or plain holes, depending on the design; it should be noted that if vanes 25 are curved as shown in FIG. 3, and with sufficient curvature forward, the openings 16 may be plain holes. If vanes 25 are straight and radial, then openings 16 should be nozzles. However, by combining curved vanes, adjusting vane lengths, using nozzles or using plain openings, suitable pressure conditions may be obtained to assure that the first fluid will flow in the desired direction, with least

amount of work required to rotate said rotor from external sources. Functionally, it is desired to obtain a high pressure and a high temperature for the said first fluid in the area where said heat exchanger means 29 is located, so that the second fluid leaving said heat exchanger means 29 is at a high temperature, and then reduce the pressure of said first fluid with most work recovery, so that said work input to said rotor is at a minimum.

The main application of the multistage heat exchanger of this invention is in steam generation; normally, the second fluid is not vaporized within the rotor but is maintained under a sufficient pressure to prevent such vaporization. Other applications include other uses where high temperature heat source is required, or where the third fluid temperature is low thus requiring a high temperature differential between said second fluid and said third fluid.

I claim:

1. A multistage heat exchanger comprising:

a. a means for supporting a shaft in bearings for rotation;

b. a shaft;

c. a rotor mounted on said shaft so as to rotate in unison therewith, said rotor having first radially outward extending passages connecting at their outward ends with first inward extending passages, said first inward extending passages connecting at their inward ends with second outward extending passages, said second outward extending passages connecting at their outward ends with second inward extending passages, said second inward extending passages connecting at their inward ends with third outward extending passages, said third outward extending passages connecting at their outward ends with third inward extending passages, said third inward extending passages connecting at their inward ends with the inward ends of said first outward extending passages for passing a first fluid, said second outward extending passage having a first heat exchanger means for removing heat from said first fluid during and after compression of said first fluid by said rotor, said third inward extending passages having second heat exchanger means for adding heat into said first fluid after said first fluid has been cooled by expansion within said rotor, said first heat exchanger means being supplied with a second fluid and circulated in heat exchange relationship with said first fluid, said second heat exchanger means being supplied with a third fluid and circulated in heat exchange relationship with said first fluid, said second fluid and said third fluid being passed into their respective heat exchangers via fluid passages within said rotor.

2. The multistage heat exchanger of claim 1 wherein said first outward extending passages are provided with nozzles at their said outward ends for passing said first fluid into said first inward extending passages.

* * * * *

[54] GAS COMPRESSOR-EXPANDER

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee,
Houston, Tex. 77042

[22] Filed: Mar. 7, 1974

[21] Appl. No.: 448,818

Related U.S. Application Data

[63] Continuation-in-part of Ser. Nos. 399,199, Sept. 20,
1973, abandoned, and Ser. No. 405,584, Oct. 11,
1973.[52] U.S. Cl. 62/401; 126/247; 165/86;
415/1; 415/64; 415/178; 416/96 R[51] Int. Cl.² F25B 3/00[58] Field of Search 62/86, 87, 401, 402, 499;
122/26; 126/247; 165/86, 88; 415/1, 64, 114,
177, 178, 179, 199 A; 416/95, 96

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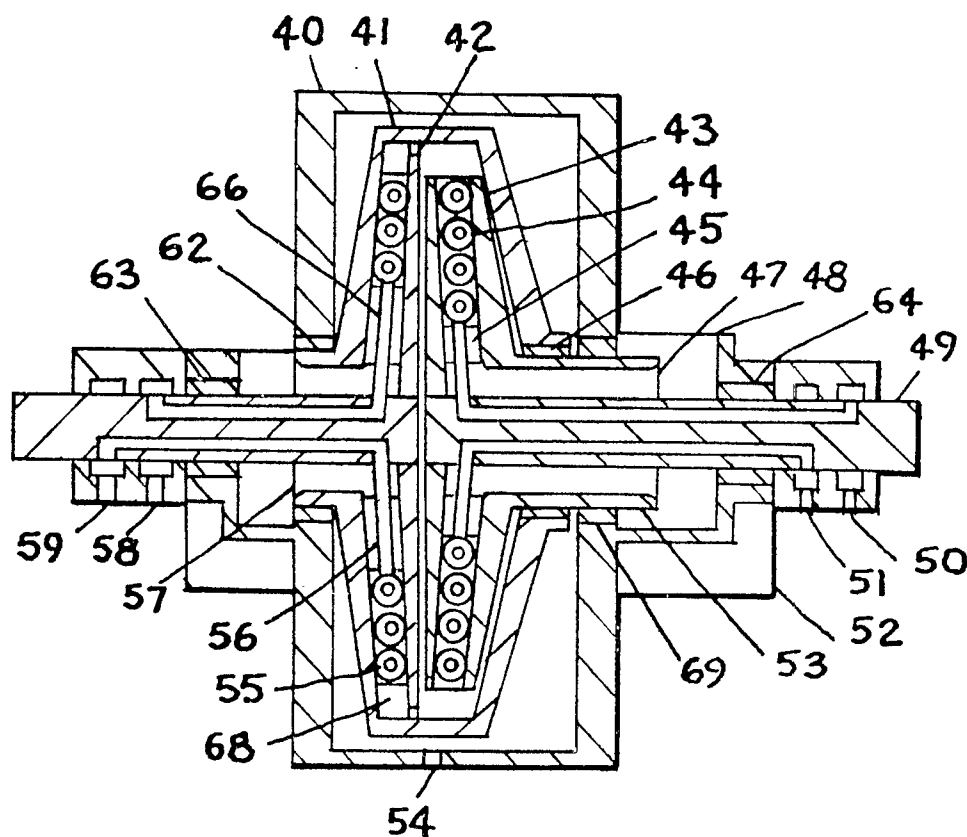
Primary Examiner—Albert W. Davis, Jr.

Assistant Examiner—Sheldon Richter

[57] ABSTRACT

A method and apparatus for compressing and expanding a fluid by passing said fluid through a rotating continuous flow centrifuge wherein said fluid is pressurized by centrifugal action on said fluid by said centrifuge rotor. Said rotor is provided with passageways for said fluid with vanes placed therewithin assuring that the fluid will rotate with said rotor. After compression, said fluid is passed in compressed state through nozzles near the periphery of said rotor with said nozzles oriented to discharge said fluid backward thus reducing the absolute tangential velocity of said fluid. After passing said nozzles, said fluid is passed through inward extending passageways to exit near the rotor center. Cooling is provided for said fluid during said compression, and heating is provided during expansion in said inward extending passages. As an alternate, two rotors may be employed, wherein said fluid is passed to a second rotor for said expansion and for deceleration. Work is required by said fluid during said compression and acceleration, and work is produced by said fluid during said expansion.

1 Claim, 5 Drawing Figures



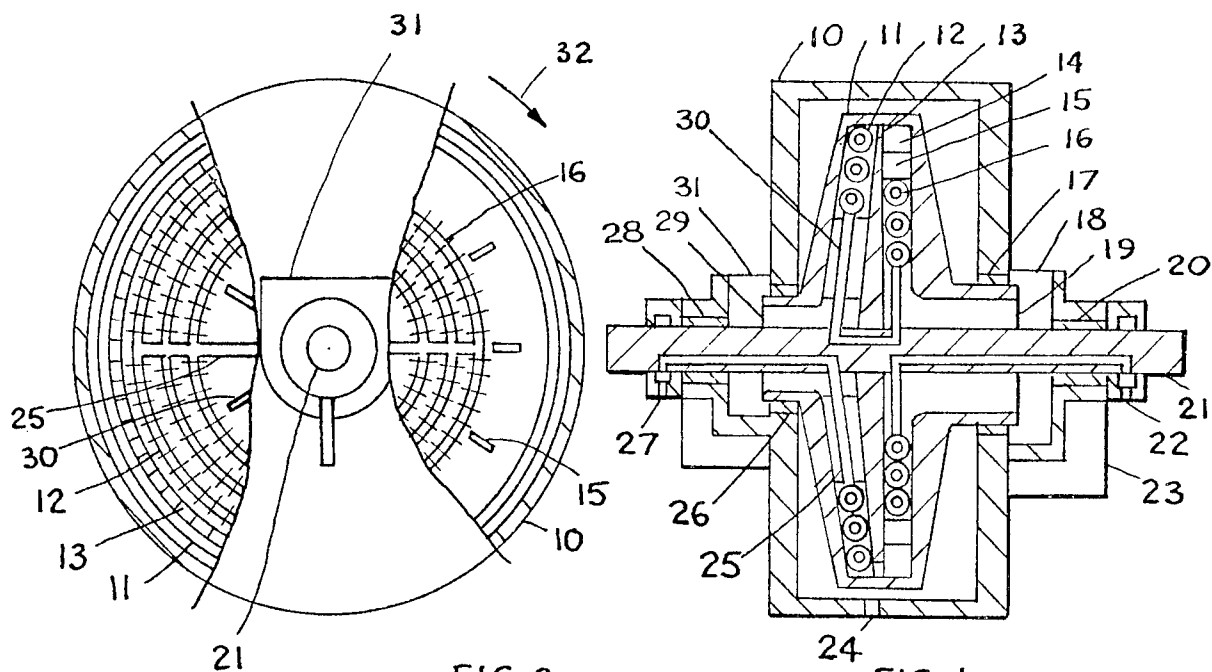


FIG. 2

FIG. 1

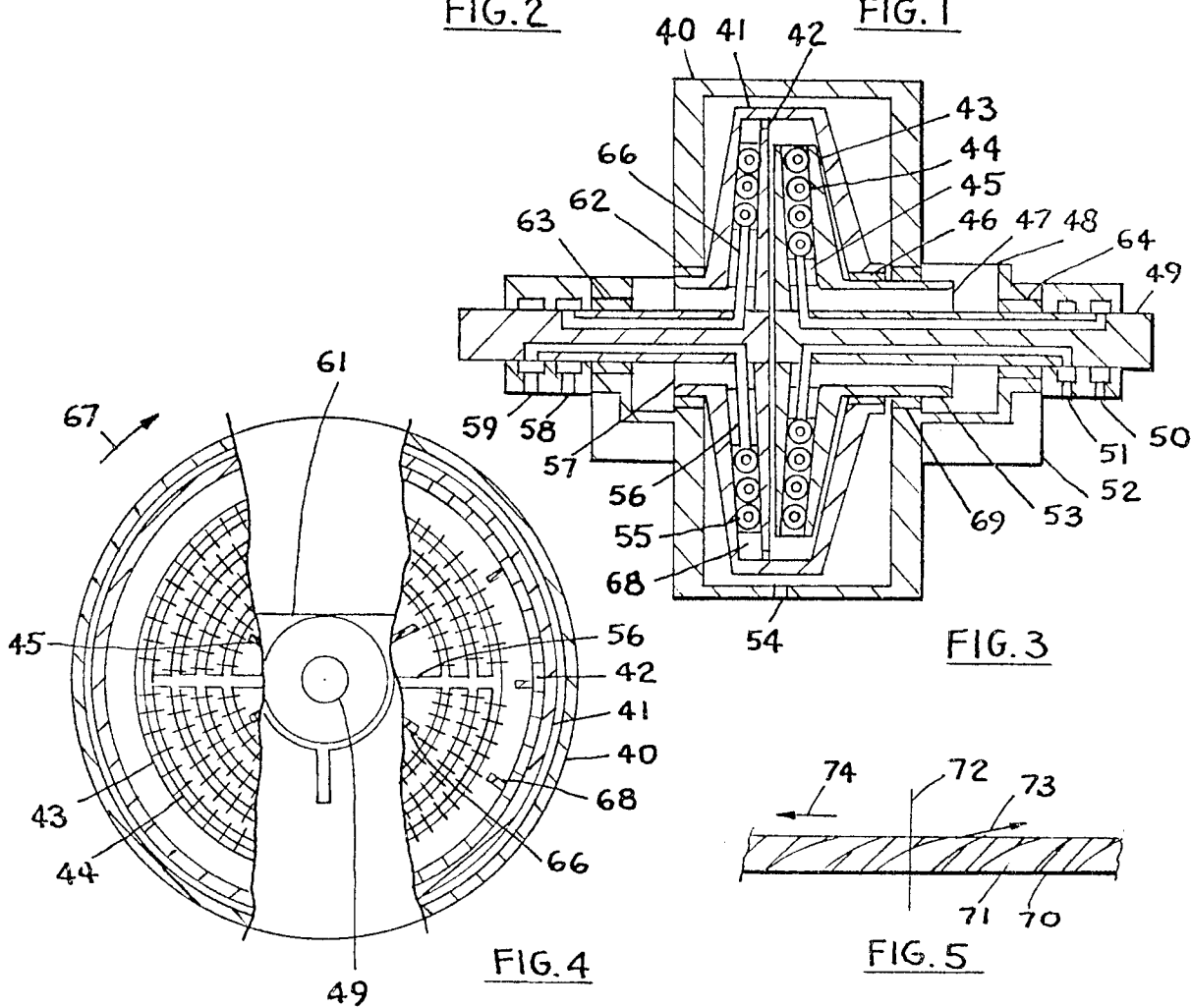


FIG. 4

FIG. 5

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GAS COMPRESSOR-EXPANDER

This application is a continuation-in-part application of "Rotary Pressurizer," filed 9-20-73, Ser. No. 399,199, now abandoned and "Compressing Centrifuge," filed 10-11-73, Ser. No. 405,584.

This invention relates to devices for compressing and expanding gases from a lower pressure to a higher pressure by employing a continuous flow centrifuge with outward extending passages at the compressing section and then having inward extending fluid passages for expansion and for deceleration of said fluid.

In my previous U.S. Pat. Nos. 3,761,195 and 3,793,848, I described similar compressors; the compressor-expander of this invention adds some improvements intended to reduce the work input to the compressor-expander.

FIG. 1 is a cross section of one form of the compressor, and FIG. 2 is an end view of the unit shown in FIG. 1.

FIG. 3 is a cross section of another form of the compressor, and FIG. 4 is an end view of the unit shown in FIG. 3.

FIG. 5 is a detail of the rotor nozzles.

It is an object of this invention to provide a means for adding heat to the fluid which is the first fluid, during expansion and deceleration within the centrifuge, for reduced work input to said compressor-expander.

Referring to FIG. 1, therein is shown a cross section of one form of the compressor-expander. 10 is casing, 11 is rotor, 12 is heat removal heat exchanger, 13 are rotor nozzles oriented to discharge said first fluid backward away from the direction of rotation, 14 is first fluid space, 15 are vanes within the first fluid expansion passage, 16 is heat addition heat exchanger, 17 and 26 are rotor seals, 18 and 19 are first fluid exits, 20 and 28 are shaft bearings, 21 is rotor shaft, 22 and 27 are second fluid inlet and outlet, 23 is bearing support, 24 is casing vent into which a vacuum pump may be connected, 25 is second fluid distribution conduit, 29 and 31 are first fluid entries, 30 is a vane within outward first fluid passage.

In FIG. 2, an end view of the unit shown in FIG. 1, is illustrated. 10 is casing, 15 is a vane, 16 is heat addition heat exchanger, 32 indicates direction of rotation, 31 is first fluid inlet, 25 is second fluid conduit, 30 is vane, 12 is heat removal heat exchanger, 13 are rotor nozzles, 11 is rotor, and 21 is rotor shaft.

In FIG. 3, another form of the compressor-expander is shown, in cross section. 40 is casing, 41 is first rotor, 42 are first rotor nozzles, 43 is second rotor, 44 is heat addition heat exchanger, 45 is vane, 46 is a bearing and seal, 47 and 48 are fluid exits, 64 and 63 are bearings and seals, 49 is first rotor shaft, 50 and 51 are third fluid entry and exit, 52 is bearing support, 53 is second rotor shaft supported by bearing and seal 68, 54 is casing vent, into which a vacuum pump may be connected, 68 is a vane, 55 is heat removal heat exchanger where second fluid is being circulated in heat exchange relationship with said first fluid, 56 is second fluid distribution conduit, 57 and 61 are first fluid inlets, 58 and 59 are second fluid inlet and outlet, 62 is rotor seal, and 66 are vanes.

In FIG. 4, an end view of the unit shown in FIG. 3 is illustrated. 40 is casing, 41 is first rotor, 42 are first rotor nozzles, 56 is second fluid conduit, 61 is first fluid entry, 67 indicates direction of rotation for both rotors,

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43 is second rotor, 44 is heat addition heat exchanger wherein said third fluid is circulated in heat exchange relationship with said first fluid, 45 are vanes, 49 is shaft, 66 are vanes.

In FIG. 5, a detail of the rotor nozzles is shown. 70 is rotor wall into which nozzles 71 are mounted, 72 indicates orientation of rotor shaft about which said rotor rotates in direction indicated by 74, and 73 indicates first fluid leaving nozzles 71.

In operation, and referring to FIG. 1, first fluid enters via opening 31 and enters rotor 11 via opening 29, and is then compressed within rotor outward extending passageways with vanes and heat exchanger fins assuring that said first fluid will rotate with said rotor. During said compression, heat is removed from said first fluid in heat exchanger 12, wherein a second fluid is circulated in heat exchange relationship with said first fluid. After said compression, said first fluid is discharged from nozzles 13 backward in a direction that is away from the direction of rotation thus reducing the absolute tangential velocity of said first fluid. Said first fluid is then decelerated and expanded in inward extending first fluid passageways with vanes and heat exchanger fins assuring that said first fluid will rotate with said rotor for recovery of work associated with said deceleration of said first fluid. After said deceleration, first fluid is discharged via opening 19, and opening 18.

The operation of the unit shown in FIG. 3 is similar to that described for the unit of FIG. 1. After discharge from the nozzles 42, the first fluid is passed to a second rotor inward extending passageways for deceleration and expansion and for recovery of work associated with said fluid deceleration. After said deceleration, said first fluid is passed to exit opening 47. Heat is added to said first fluid in heat exchanger 44 wherein a third fluid is being circulated in heat exchange relationship with said first fluid.

In the unit of FIG. 1, heat is added to said first fluid in heat exchanger 16, where said second fluid may be circulated in heat exchange relationship with said first fluid; usually, said second fluid is first passed through said heat removal heat exchanger 12, and then passed to said heat addition heat exchanger 16.

In the unit of FIG. 3, said second fluid may be passed to heat exchanger 44 and used as said third fluid. Also, in the unit of FIG. 1, a third fluid may be employed in heat exchanger 16, as desired.

The work input to the compressor of this invention is the difference between the work used to accelerate said first fluid to the rotor tangential velocity, and the work recovered from said first fluid during said expansion and deceleration. Heat removal during compression and acceleration reduces the first fluid density and thus results in a higher first fluid pressure at rotor periphery for a predetermined rotor rotational speed. Heat addition during expansion and deceleration will result in a reduced first fluid density, and thus a lesser pressure loss in said deceleration, for a predetermined rotor tangential speed. Normally, said compression of said first fluid is nearly isothermal, depending of second fluid entry temperature, and similarly, the expansion of said first fluid may be isothermal, depending of the temperature of the heat addition fluid in heat exchangers 44 and 16. Thus, very low work input to said compressor results.

Said second fluid and said third fluid are passed to said heat exchangers through passageways within said

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rotor shafts, and thus the work requirement for said fluids is nearly nil.

The first fluid is normally gas, being compressed to a higher pressure, such as air. Said second fluid and said third fluid are usually liquids, such as water, but gases also can be used if desired.

Various gauges and governors may be used with the compressor of this invention. They do not form a part of this invention and are not further described herein.

The rotating components of the compressor are normally made of high strength materials, usually metals. The heat exchangers are shown to be made of finned tubing, but other materials may be used, or the heat exchangers built to rotor walls. Vanes are provided to assure that the first fluid will rotate with the rotor, and the heat exchanger fins also serve as said vanes.

The unit shown in FIG. 1 has a single rotor, with the first fluid being discharged from nozzles 13 backward, and then the vane 15 tips are intended to have approximately the same tangential velocity as the first fluid entering the spaces between said vanes 15. Many fluids, especially gases with high viscosity, may be accelerated by the rotor 11 in space 14, thus making it difficult to obtain the desired tangential velocity reduction for said first fluid. This difficulty is provided for in the unit of FIG. 3, where a second rotor may rotate at any desired

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speed, thus assuring that the first fluid tangential velocity after leaving nozzles 42 is maintained at a value required for best performance.

I claim:

1. In a compressor-expander having two rotating rotors with a fluid to be compressed being the first fluid, with a first rotor having an entry for said first fluid near the center of rotation with said first fluid then being accelerated and compressed in outward extending first rotor passages and said first fluid then being passed into a second rotor for deceleration and expansion, and to exit near the center of rotation of said second rotor, with heat being removed from said first fluid in a heat removal heat exchanger within said first rotor outward passages by circulating a second fluid in heat exchange relationship with said first fluid, the improvement comprising:

a. a heat addition heat exchanger within said inward extending second rotor first fluid passages for adding heat to said first fluid during said expansion and deceleration, by circulating a third fluid in heat exchange relationship with said first fluid, with a third fluid being supplied and discharged via passages carried by said second rotor.

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[54] **HEAT EXCHANGER**

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee,
Houston, Tex. 77042

[22] Filed: **Apr. 3, 1975**

[21] Appl. No.: **564,615**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 393,571, Aug. 31,
1973.

[52] **U.S. Cl.**..... **62/499**; 62/86;
62/401; 122/11; 165/88; 415/1; 415/178

[51] **Int. Cl.²**..... **F25B 3/00**; F25D 9/00

[58] **Field of Search**..... 62/86, 87, 401, 402,
62/499-502; 122/11, 26; 126/247; 165/86,
88, 105; 415/1, 114, 177-179; 416/95, 96

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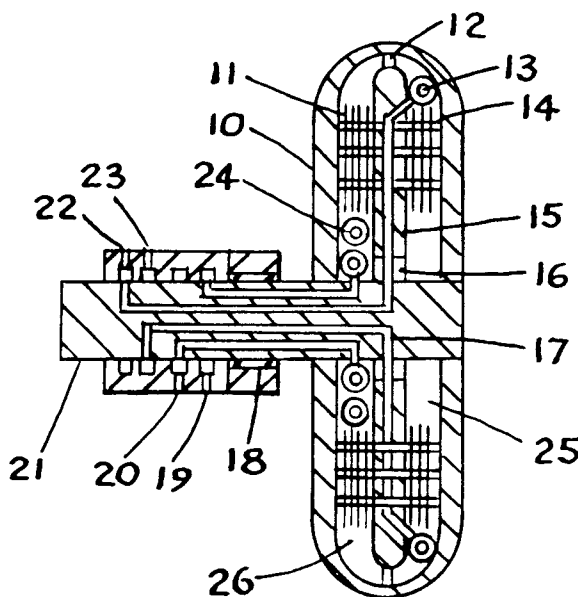
Primary Examiner—Albert W. Davis, Jr.

Assistant Examiner—Sheldon Richter

[57] **ABSTRACT**

A means for exchanging heat between two streams of a working fluid, during expansion and compression of said working fluid, thus providing heat removal from said working fluid either during expansion or compression and providing heat addition to said working fluid either during expansion or compression. The heat exchanger means may be either a finned wall, finned tubing, or heat pipes; also, other types of heat exchangers may be used. The heat exchangers are normally mounted within a centrifuge type rotor wherein said working fluid is either cooled or heated when said working fluid either expands or is being compressed within outward extending rotor passages.

3 Claims, 4 Drawing Figures



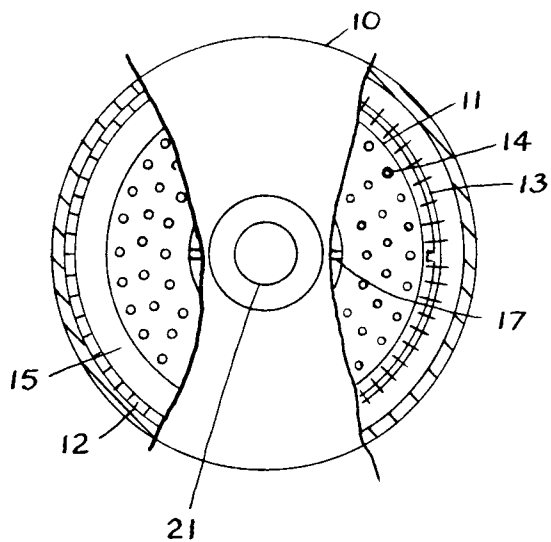


FIG. 2

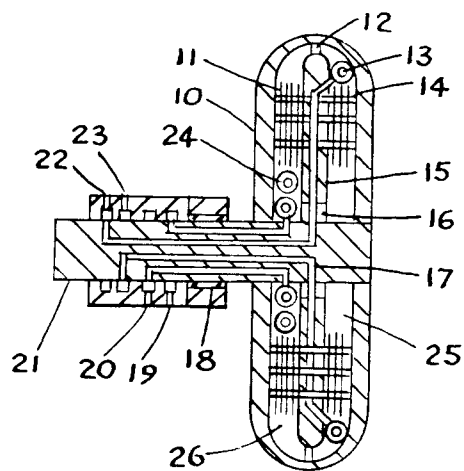


FIG. 1

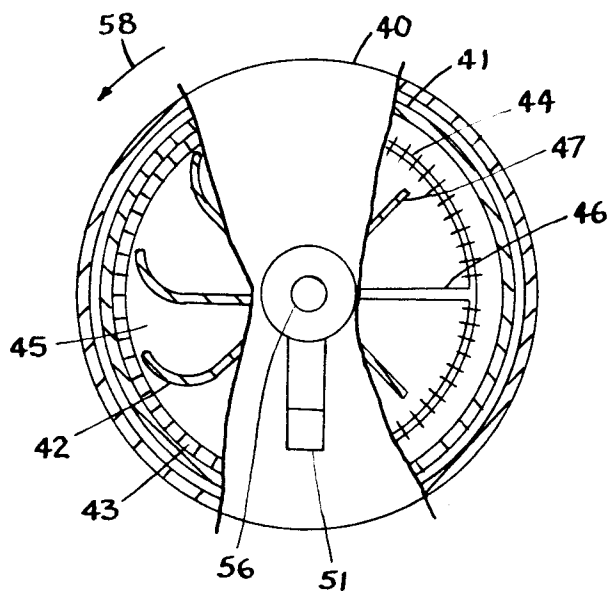


FIG. 4

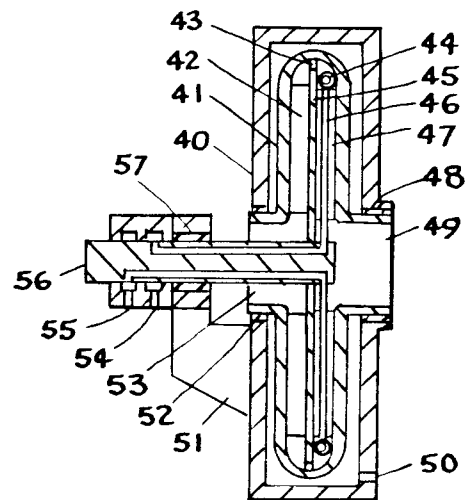


FIG. 3

HEAT EXCHANGER

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuing-in-part application of "Rotary Heat Exchanger", Ser. No. 393,571, filed 8-31-73, pending 3/15/76.

BACKGROUND OF THE INVENTION

This invention relates to compressors, turbines and heat exchangers of the rotary type wherein a heat exchanger is provided to transfer heat between two streams of the working fluid while said working fluid is within the rotor of the device.

Various types of rotary fluid handling devices exist, and usually the working fluid is either heated or cooled while said working fluid is within the rotor, but said cooling or heating is provided by circulating a separate heating fluid or cooling fluid from external sources into the rotor heat exchangers. One such device is shown in my previous U.S. Pat. No. 3,793,848.

It is an object of this invention to provide a method and apparatus for the transfer of heat into or out of said working fluid during expansion or compression while within the rotor by using another stream of the same working fluid within the rotor as the heat source or heat sink. Thus, for example, heat is transferred from the said working fluid during compression, into another stream of the same working fluid during expansion, thus providing cooling for the fluid stream being compressed, and heating for the fluid stream being expanded; therefore, as is well known, the work of compression for said fluid will be decreased, and work of expansion for said working fluid will be increased, thus resulting in a lower overall work required for the rotor. Similarly, the method can be used, in reverse, for turbines, where heat is taken from the working fluid during expansion, and added during compression. Thus, to provide such cooling and heating for said working fluid, no external heat or cooling source is required, for that part of the process, resulting in a simplified and thus more economical rotor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 illustrates a cross section of a closed cycle rotor, and

FIG. 2 is an end view of the rotor shown in FIG. 1.

FIG. 3 is an illustration of another rotor, of the open type, and

FIG. 4 is an end view of the unit of FIG. 3.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, therein is shown one form of the invention applied to a closed type centrifuge rotor. In this rotor, a working fluid is sealed within rotor and circulates therewithin with compression on one side of rotor and expansion on another side of the same rotor, with finned tube type heat supply and heat removal heat exchangers, and a heat pipe type heat exchanger, one embodiment of this invention, providing heat transfer from the working fluid stream being compressed into another stream of the same working fluid being expanded. The rotor 10 is supported by shaft 21 and bearing 18, with working fluid being compressed on side 25 with heat pipes 14 and fins 11 removing heat during compression, with said working fluid then passing through finned tube type heat exchanger 13 where

heat is transferred into a second fluid being circulated within said heat exchanger 13, and after that said working fluid passes through openings 12, which may be nozzles, into expansion side 26 where said working fluid while traveling toward rotor center expands, and heat is being added into said working fluid by the heat exchanger formed by said heat pipes 14 and said fins 11. Said heat exchanger transferring heat from one stream of working fluid to another stream of working fluid will be hereinafter called the working fluid heat exchanger. During and after said expansion, heat is added into said working fluid from a third fluid being circulated in heat exchanger 24, and then said working fluid is passed through opening 16 to be again compressed thus completing its cycle. 22 and 23 are second fluid entry and exit, and 19 and 20 are third fluid entry and exit, 15 is rotor divider wall, 17 is second fluid distribution conduit.

In FIG. 2, an end view of the unit of FIG. 1 is shown. 10 is rotor, 11 are fins, 14 are heat pipes, 13 is a second fluid heat exchanger, 17 are second fluid conduits, 21 is shaft, 12 are working fluid passages, and 15 is rotor center wall.

In FIG. 3, another form of the working fluid heat exchanger is shown, mounted within an open cycle rotor. In this form said working fluid heat exchanger consists of fins, serving as vanes also, mounted on a heat conductive center wall, transferring heat from one stream into another working fluid stream. 40 is rotor casing supporting seals 48 and 52, shaft 56 and shaft bearing 57. 41 is rotor, 42 is a fin, 43 are openings for the working fluid that may also be nozzles, 44 is second fluid heat exchanger that may be omitted in some applications, 45 is rotor center wall into which fins 42 and 47 are mounted with said wall 45 being heat conductive, 46 is second fluid conduit, 49 is working fluid inlet and 53 is working fluid outlet, 50 is a casing vent into which a vacuum source may be connected, 51 is bearing bracket, 54 and 55 are second fluid inlet and outlet.

In FIG. 4, an end view of the unit of FIG. 3 is shown. 40 is casing, 41 is rotor, 44 is second fluid heat exchanger, 47 are fins serving also as vanes, 46 are second fluid conduits, 51 is support for bearing, 56 is shaft, 43 are working fluid passages, 42 are fins serving also as vanes, 45 is center wall, and 58 indicates direction of rotation for the rotor.

In operation, a working fluid is compressed by centrifugal force with accompanying temperature increase, and then expanded with accompanying temperature decrease. Heat is transferred by said working fluid heat exchanger from one stream of said working fluid into another stream of said working fluid, while said working fluid is being either compressed or expanded. The direction of heat transfer depends of the use of the rotor, wherein said working fluid heat exchanger is mounted, as described hereinbefore.

In the drawings, two forms of the working fluid heat exchanger are shown, one using heat pipes, and another using heat conductive center wall and fins. Other forms of heat exchange can be used, such as finned tubing wherein a fourth fluid is circulated to transport the heat from one side of the center wall to another side. Also, said fourth fluid can be circulated to outside the rotor if desired. Also, the working fluid heat exchanger may be made tapered rather than radial in shape, if desired, so as to increase the heat transfer area of the working fluid heat exchanger, and also to provide controlled pressure changes for said working fluid

during heat transfer. Such modifications of the working fluid heat exchanger depend of the fluid being used, and also of the process variables that the said working fluid passing through the heat exchanger must follow.

In FIG. 4, the fins 42 are shown curved. Fins 47 also may be made in curved shape, if desired, to provide for circulation or pressure differentials for said working fluid within said rotor. Similarly, openings 43 may be made into nozzles, to discharge either forward or backward as desired, to provide for desired functions for said working fluid. Further, the fins 42, and fins 47 and heat exchanger 44 fins may be made of differing radial lengths to provide for suitable pressure differentials in a conventional manner.

The rotor of FIG. 1 may be mounted within a casing if desired, or suitable friction discs may be mounted outside the rotor to reduce fluid friction on rotor outer surfaces.

The heat exchanger of the unit of FIG. 1 is normally made by using disc shaped fins made of heat conductive material such as aluminum, and which are normally self supporting against centrifugal stresses, and heat pipes partially filled with liquid and partially filled with gas heat transfer fluid of suitable volatility so that on the hot side the heat transfer fluid is vaporized and on cold side the heat transfer fluid is liquefied, thus providing heat transfer by vapor. Alternatively, the pipes may be filled with a liquid or a gas, which under the centrifugal force of the rotating rotor will circulate due to density changes brought by temperature differences. The heat pipes are usually mounted snugly into the circular disc fins and are supported by said fins, and also by the rotor walls.

The heat exchanger 44 shown in FIG. 3 may be omitted if desired, especially in situations where the rotor is used as a compressor. This omission will simplify the construction of the rotor, while still providing for the working fluid compression with cooling, and expansion with heating, and thus reducing work input to rotor.

It should be noted that the types of working fluid heat exchangers used in the rotors of FIG. 1 and FIG. 3 may be exchanged with each other, as desired. Generally, the type of working fluid heat exchanger shown in FIG. 1 is used where the amount of working fluid flowing is large, since it is relatively easy to build a large surface heat exchanger of the type shown in FIG. 1. The working fluid heat exchanger of FIG. 3 is more suited for small applications, such as small air compressors.

I claim:

1. In a rotor wherein a working fluid is pressurized by centrifugal force in a set of outward extending passageways, and wherein said working fluid is expanded within a set of inward extending passageways with an accompanying pressure decrease, with said outward extending passageways extending away from a center of rotation of said rotor, and with said inward extending passageways extending inward toward the center of rotation of said rotor, with said outward extending and said inward extending passageways being in communication via passages at their outward ends to allow said working fluid to circulate, the improvement comprising a working fluid heat exchanger placed within said rotor to be in heat exchange relationship with said working fluid while said working fluid is being pressurized within said outward extending passageways, and to be in heat exchange relationship with said working fluid while said working fluid is being expanded within said inward extending passageways thus providing heat exchange between said working fluid being pressurized and said working fluid being compressed.

2. The working fluid heat exchanger of claim 1 wherein said heat exchanger comprises at least one conduit filled with a heat transfer fluid.

3. The working fluid heat exchanger of claim 1 wherein said heat exchanger comprises a heat conductive wall between said working fluid being compressed and said working fluid being expanded, within said outward extending and said inward extending working fluid passageways.

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[54] **HEAT EXCHANGER**

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee, Houston, Tex. 77042

[22] Filed: **Dec. 2, 1974**

[21] Appl. No.: **528,878**

Related U.S. Application Data

[62] Division of Ser. No. 423,560, Dec. 10, 1973, abandoned, which is a division of Ser. No. 357,588, May 7, 1973, abandoned.

[52] U.S. Cl. **62/87; 62/499; 165/2; 165/88; 415/1**

[51] Int. Cl.² **F25B 3/00**

[58] Field of Search **62/499, 86, 87; 415/147, 1; 165/1, 2, 86, 88**

Primary Examiner—**Albert W. Davis, Jr.**

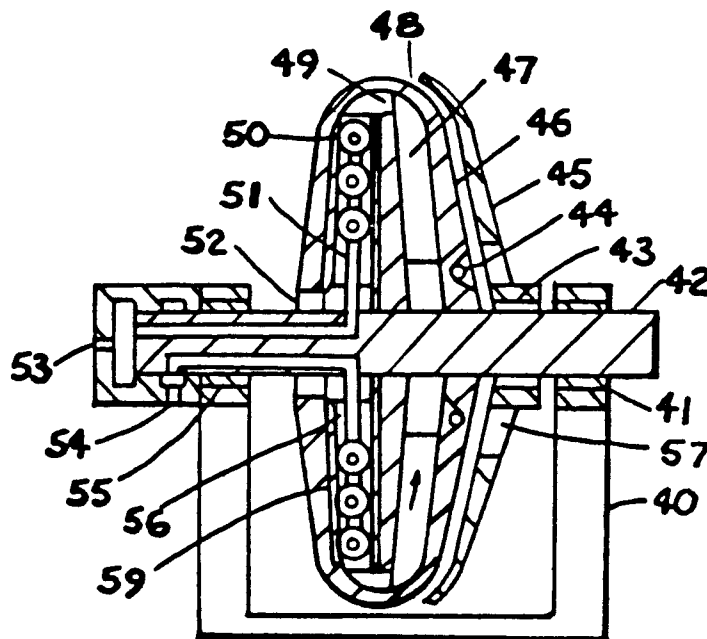
Assistant Examiner—**Sheldon Richter**

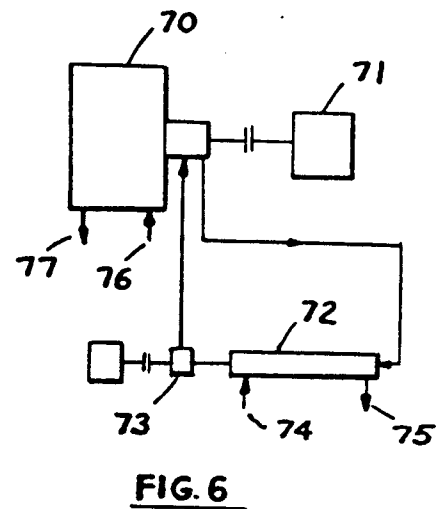
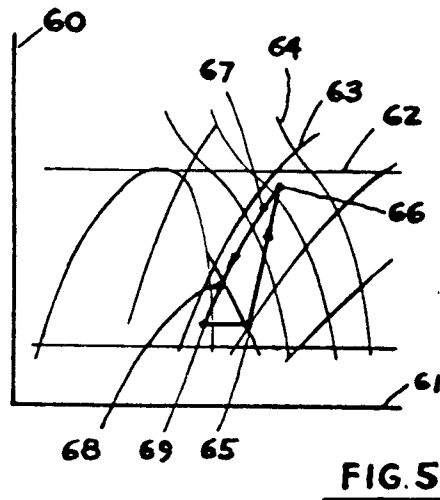
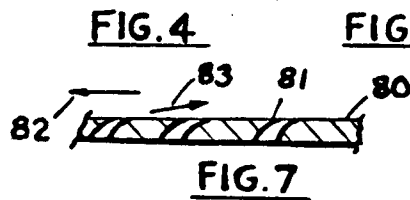
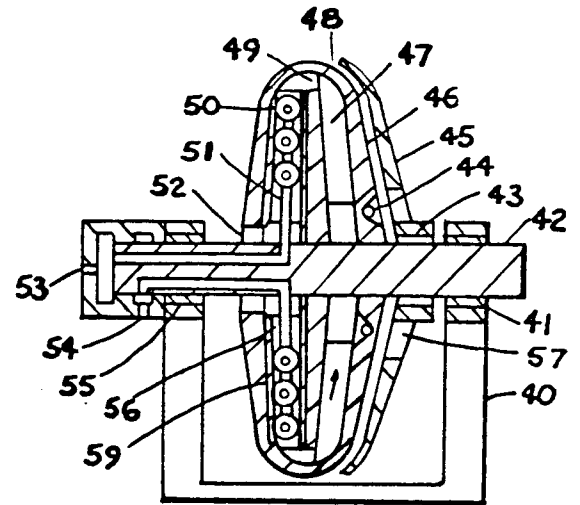
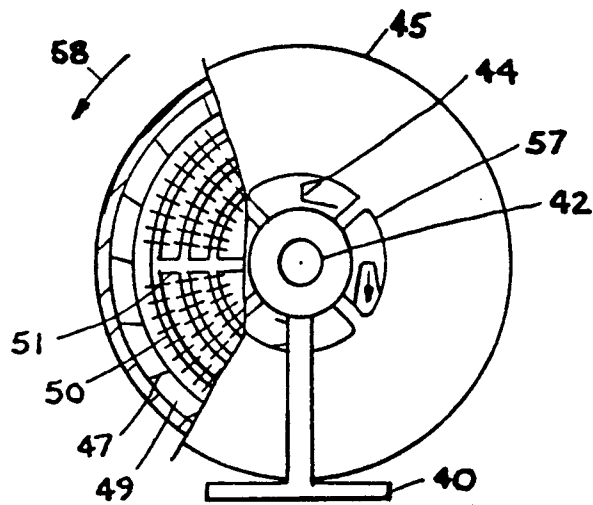
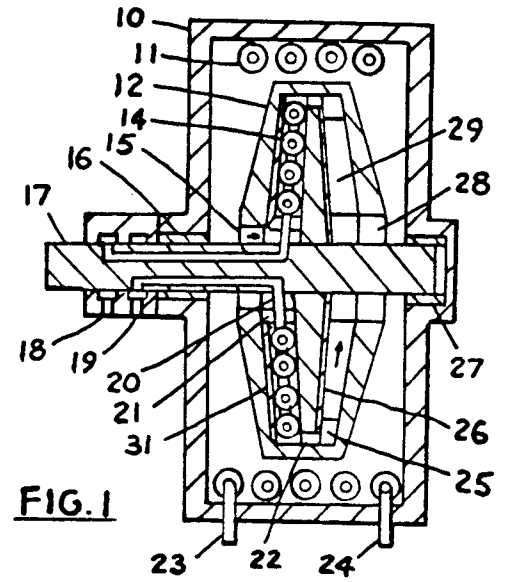
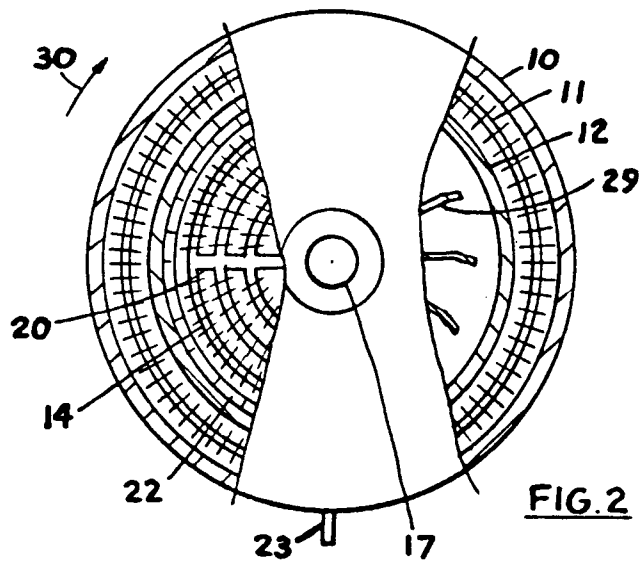
[57] **ABSTRACT**

A method and apparatus for transferring heat from a

first fluid which is normally gaseous to a second fluid which is normally a liquid, by using a rotating rotor with passages for said first fluid extending from rotor center outward and with passages for said second fluid also extending outward within said rotor with said two fluids being in heat exchange relationship within said rotor with heat another transferred from said first fluid to said second fluid, wherein said first fluid temperature is increased by compressing said first fluid within said rotor. Said two fluids are then returned in separate passages to the center of said rotor and discharged; said first fluid will leave said rotor colder than it entered and said second fluid will leave said rotor warmer than it entered. In one form of the invention, said rotor is mounted within a sealed casing with entry and exit for first fluid to said casing, and a heat exchanger for adding heat is provided. Also, in another form of the invention, nozzles are added to rotor walls for discharging said first fluid backward thus recovering some work. Also, a means is included for reducing external friction on rotor.

1 Claim, 7 Drawing Figures





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HEAT EXCHANGER

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a divisional patent application of "HEAT EXCHANGER", filed 12/10/73, Ser. No. 423,560 now abandoned. This patent application is also a divisional patent application of "Steam Generator", filed 5/7/73, Ser. No. 357,588 now abandoned.

This invention relates generally to means for compressing a gaseous first fluid with accompanying temperature increase and transferring heat from said gaseous first fluid to a second fluid during and after said compression, thus producing a heated second fluid and a cooled first fluid.

The art of heating and cooling has seen a variety of devices. In some of these devices, such as heat pumps, where a fluid is compressed in a compressor, then heat is removed in a heat exchanger after which the pressure is reduced, and then heat is added to the fluid, heating and cooling is produced by said two heat exchangers and work is supplied into said compressor. The main disadvantage is that these devices will require large amounts of power input to produce said heating and cooling.

In FIG. 1, a cross section of one form of the heat exchanger is shown, and in

FIG. 2, an end view of the unit shown in FIG. 1 is illustrated.

In FIG. 3, a cross section of another form of the heat exchanger is shown, and in

FIG. 4 an end view of the unit shown in FIG. 3 is illustrated.

In FIG. 5, a typical pressure-enthalpy diagram for the first fluid used for the heat exchanger is shown with a work cycle illustrated thereon.

In FIG. 6, a typical schematic diagram for the application of the heat exchanger is shown.

In FIG. 7, a detail of nozzles is illustrated.

It is an object of this invention to provide a means for producing heating by employing low temperature heat sources, and doing it in compact and relatively low cost device. Also, it is an object of this invention to provide a heat exchanger wherein the work input is low thus providing an economical means for producing said heating. Further, it is an object of this invention to provide a means for producing cooling.

Referring to FIG. 1, therein is shown a cross section of one form of the heat exchanger. 10 is casing, 11 is stationary heat exchanger for adding heat to said first fluid, 12 is rotor, 14 is rotor heat exchanger for said second fluid, 15 is first fluid entry to said rotor, 16 is shaft bearing and seal, 17 is rotor shaft, 18 and 19 are second fluid entry and exit to said rotor, 20 is second fluid distribution conduit, 21 is vane, 22 is first fluid nozzles, 23 and 24 are heating fluid entry and exit, 25 is first fluid space, 26 is thermal insulation, 31 is thermal insulation layer, 27 are bearings, 28 is rotor first fluid exit, 29 is vane on rotor exit side.

In FIG. 2, an end view with portions removed, is shown of the unit illustrated in FIG. 1. 10 is casing, 11 is heat exchanger, 12 is rotor, 29 are vanes on exit side of rotor, 17 is rotor shaft, 23 is heating fluid conduit, 22 are rotor first fluid internal nozzles, 14 is heat exchanger, 20 is second fluid conduit, and 30 indicates direction of rotation for rotor.

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In FIG. 3, another form of the heat exchanger is shown in cross section. 40 is support base, 41 is shaft bearing, 42 is rotor shaft, 43 is friction reducer bearing, 57 is friction reducer opening to allow discharge of first fluid, 44 is first fluid discharge nozzle from rotor, 45 is friction reducer disc, 46 is rotor, 47 is rotor vane on rotor discharge side, 48 indicates space between rotor and friction reducer disc, 49 is first fluid passage at rotor periphery, 50 is rotor heat exchanger, 51 is second fluid conduit, 52 is first fluid inlet to rotor, 53 is second fluid inlet to rotor shaft passage, 54 is second fluid discharge, 55 is shaft bearing, 56 is vane, 59 is thermal insulation layer.

In FIG. 4, an end view of the unit shown in FIG. 3 is shown, with portions removed to illustrate interior details. 45 is friction reducer disc, 44 are first fluid exit nozzles, 57 are openings in friction reducer disc to allow first fluid discharge from rotor, 42 is rotor shaft, 40 is base, 49 is fluid passage with vanes 47, 50 is rotor heat exchanger for transferring heat from first fluid to second fluid, 51 are second fluid conduits, and 58 indicates direction of rotation for rotor.

In FIG. 5, a pressure-enthalpy diagram for a typical first fluid is shown. 60 is pressure line and 61 is enthalpy line, 62 are constant pressure lines, 63 are constant entropy lines and 64 are constant temperature lines. For the unit of FIG. 1, the cycle within rotor with external heat addition is 65 to 66 compression with cooling, 66 to 67 expansion which is isentropic, in nozzles 22 at rotor periphery, and 67 to 69 expansion in rotor vanes, with heat addition from 69 to 65. For the unit shown in FIG. 3, the compression is from 65 to 66, expansion in rotor exit side vanes from 66 to 68, and expansion in rotor exit nozzles from 68 to 69. If the rotor of FIG. 3 was within a closed casing, then heat addition would be from 69 to 65, same as for the unit of FIG. 1.

In FIG. 6, a typical schematic diagram is shown to indicate typical application of said heat exchanger. 70 is heat exchanger connected to drive means 71 via a power transmission shaft, 72 is a heat exchanger to remove heat from said second fluid which is circulated via conduits from said heat exchanger, 73 is circulation pump for said second fluid, 74 is a supply of cool fluid to heat exchanger 72 and 75 is discharge of hot fluid from heat exchanger 72, 76 and 77 are heating fluid entry and exit to said heat exchanger 70 stationary heat exchanger.

In operation, power is supplied to said heat exchanger rotor shaft from an external power source, causing it to rotate. First fluid enters said rotor via entry at rotor center and is compressed by centrifugal action on said first fluid by said rotor with accompanying temperature increase. The second fluid is circulated within said rotor in separate fluid passages in heat exchange relationship with said first fluid and heat is transferred from said hot first fluid to colder second fluid. The said first fluid is then passed to inward extending exit passageways near the periphery of said rotor, and discharged from said rotor near the rotor center. The second fluid is also passed along its own passageways to exit passages located within said rotor shaft and from there to exit. The compression of said first fluid is with cooling for best efficiency, with heat being transferred during compression from first fluid to second fluid, and heat may also be transferred after compression from said first fluid to said second fluid. The expansion in the exit side vanes is normally isentro-

pic except for heat gains from rotor walls. The first fluid passageways for the rotor area for compression, is usually thermally insulated to prevent heat loss to rotor walls and to surrounding space, and also to prevent overheating said rotor structure which could lead to rotor failure.

There are two methods shown in the drawings to reduce work input to rotor, while still assuring the propelling of said first fluid through said rotor. In FIG. 1, there are shown nozzles at rotor periphery, item 22, and these nozzles are arranged to pass said first fluid backward in direction away from direction of rotation so that the first fluid absolute velocity will be less than the tangential velocity of said rotor in the area of said nozzles; this velocity reduction will provide for a suitable pressure differential to assure that said first fluid will be transported through said rotor. Said fluid will be discharged from nozzles 22 to space 25 for velocity adjustment before entry of said first fluid to exit side inward extending passageways formed by vanes 29. It should be noted that the actual exit velocity for said first fluid relative to said nozzles 22, is small, often in the area of 50-100 feet/second for most gases, and said nozzles are usually of the converging type. However, said nozzles 22 are sized and shaped to provide for maximum exit velocity for the pressure differential existing between entry and exit of said nozzles.

The second method for assuring that said first fluid will be transported through said rotor is shown in FIG. 3, where exit nozzles are provided for the first fluid leaving said rotor. Usually said exit nozzles 44 are located at a distance from rotor center that is greater than the radius of entry opening to said rotor for said fluid; this provides for needed pressure differential to assure that said first fluid passes through said rotor, while simultaneously producing thrust on said rotor wheel to reduce the work input to said rotor wheel from external sources. Said nozzles 44 are oriented to discharge said first fluid tangentially backward away from the direction of rotation, thus producing thrust. The velocities for the first fluid leaving said exit nozzles 44 are usually higher than those for nozzles 22, and nozzles 44 are sized and shaped to obtain highest attainable exit velocity for said first fluid leaving said rotor nozzles, for the pressure difference available.

It should be noted that the cooling of said first fluid during compression, as shown in FIG. 5, increases the first fluid pressure gain for the entry side of said rotor, and thus reduces the requirement for the need for other means to assure passage of said first fluid through said rotor. By cooling during compression said first fluid, and by using the other described means to reduce work input to said rotor, very low work input values to said rotor can be realized, thus providing for a very economical heat and cooling generating source.

The friction reducing disc shown in FIG. 3, item 45, is intended to reduce friction losses on rotor external surfaces especially in the areas where rotor surface speeds are high. This is done by allowing said disc 45 to rotate at its own speed; this speed can be regulated approximately to a correct value by setting the space between said rotor wall and said disc so that by gas friction said disc will rotate at approximately one half the speed of said rotor. This speed for said disc will reduce, theoretically, the rotor friction loss by approximately 40%, from the value without said disc. This is due to the lower velocity differential between rotor surface and surrounding fluid, and similarly between

said disc and surrounding fluid. In FIG. 3, one such friction reducer is shown, mounted on one side of said rotor; said discs may be mounted on both sides of said rotor, and similarly also for the rotor of FIG. 1. By this means, the friction losses can be reduced for the rotors by nearly half, thus improving the overall performance of the heat exchanger. Alternately, said friction reducer discs may be omitted, and it should be noted that for lower rotor speeds, said discs are not of a great value.

In FIG. 7, a detail of nozzles 22 is shown. 80 is rotor center wall, 81 is a nozzle, 82 indicates direction of rotation of rotor and 83 indicates fluid leaving said nozzles 81. It should be noted that the fins of heat exchanger coils 14 could be slanted backward to also add to the pressure differential to transport said first fluid through rotor; this is also illustrated in FIG. 2; with said slanting of said fins which act as vanes, said nozzles 22 could be deleted, or the two methods used in combination as shown in FIG. 2.

The heat exchanger of this invention can be used to provide heating for various applications. For lower temperature needs, air can be used as the first fluid; for high temperature requirements, fluids with lower specific heats are needed. Also, for least work input to said rotor, the fluid should have properties wherein the work input between rotor entry and periphery, and the work output between rotor periphery and exit are nearly the same, while at the same time having a low specific heat at constant pressure.

The heat exchanger of this invention can be also used to generate steam. Normally, the second fluid is maintained in its liquid state at all points by pressurizing the said fluid, and the hot second fluid is then used to generate said steam in heat exchanger 72 of FIG. 6. The heat exchanger of this invention is intended for high rotational speeds with heavy material sections especially at rotor walls near rotor center; said high speeds are required normally for steam generation.

Various governors, controls and the like are used with the device; they are not a part of this invention and are not described.

The heat exchangers within the rotors are shown to be constructed from finned tubing, attached to rotor walls for support. Other types of heat exchangers may be used to obtain the same function of transferring heat from the hot first fluid to said second fluid. Normally, the heat exchangers are arranged for parallel flow, with both the said first fluid and said second fluid progressing outward from center together, in their separate passageways and in heat exchange relationship with each other, and then said second fluid is returned back to rotor center and passed along shaft passages to discharge. For maximum heat output especially at high second fluid exit temperatures, said parallel flow for said fluids is mandatory. It should be noted, that leaving temperatures for said second fluid may be in the range of 300° to 1000°F, depending of the rotor speed and the first fluid used.

The work input to rotor shaft is low. The second fluid enters and leaves the rotor near rotor center, and thus the work input for said second fluid is nearly zero, within the rotor. The first fluid is circulated by either having a pressure differential generated at nozzles near rotor periphery, or circulation is assured by discharging said first fluid from nozzles located further away from rotor center than the furthest distance from rotor center to entry openings; this could be compared to a

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siphoning effect. Work is required to accelerate the first fluid when such first fluid is passing outward within said rotor, and work is received by said rotor when said first fluid is decelerated during passage inward toward rotor center. The first fluid is forced to travel near the speed of the rotor by having vanes or other obstructions such as heat exchanger fins, within said inward and outward first fluid passages. Thus, the work input at the rotor shaft is the difference between the work of acceleration required on the compression side of said rotor, and the work of deceleration recovered on the expansion side of the rotor; also, some work is received by the rotor in the said peripheral nozzles 22, or in discharge nozzles 44.

I claim:

1. A method of transferring heat from a lower temperature to a higher temperature comprising compressing a first fluid by a centrifugal force within an outwardly extending first passage of a rotating rotor with

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accompanying temperature increase; expanding said compressible first fluid inwardly in a second passage that is connected at its outward end with said outwardly extending passage outer end by a first passage means, connecting said second passage inward end with a discharge means a second radial distance away from the center of rotation of said rotor, connecting said first passage inward end with an entry means with said entry means furthest radial distance from the center of rotation of said rotor being less than said second radial distance to provide a pressure differential for said first fluid within said rotor, said discharge means being a set of nozzles adapted of passing said first fluid and being oriented to discharge said first fluid away from the direction of rotation to generate torque on said rotor to assist in the rotation of said rotor, removing heat from said first fluid during and after compression within said rotor in a heat exchanger carried by said rotor.

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[54] **FLUID PRESSURIZER** 3,758,223 9/1973 Eskeli 415/80
 [76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee, 3,828,553 8/1974 Eskeli 415/80
 Houston, Tex. 77042 3,879,152 4/1975 Eskeli 415/80

[22] Filed: **Apr. 17, 1975**

Primary Examiner—C. J. Husar

[21] Appl. No.: **568,895**

[57] **ABSTRACT**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 386,273, Aug. 7, 1973, Pat. No. 3,758,223, and a continuation-in-part of Ser. No. 185,060, Sept. 30, 1971, Pat. No. 3,879,152.

[52] **U.S. Cl.** **415/80**

[51] **Int. Cl.²** **F01D 1/18**

[58] **Field of Search** **415/80**

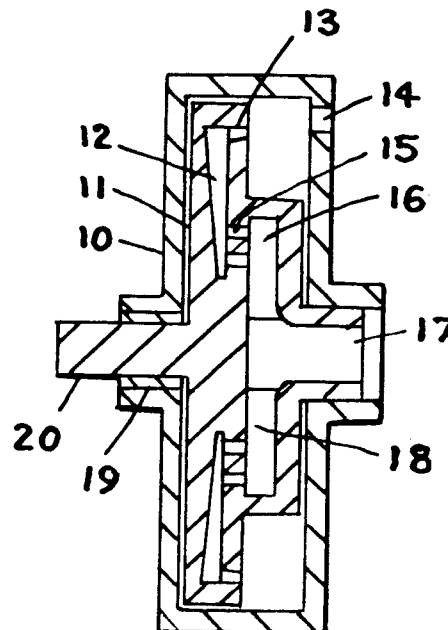
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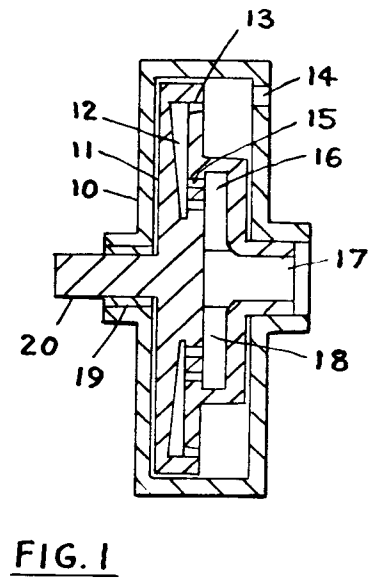
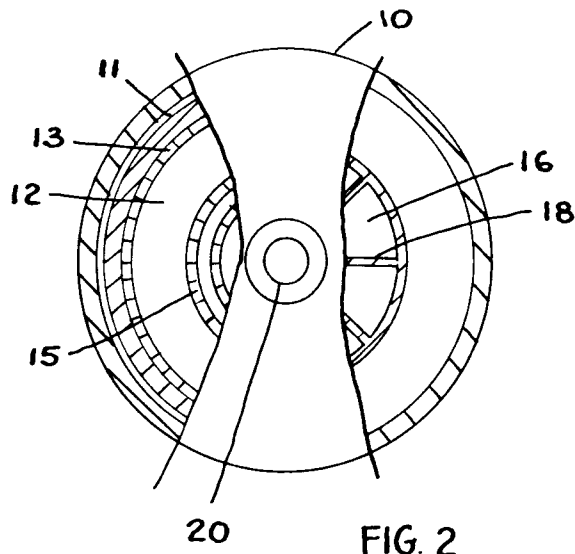
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A method and apparatus for the pressurizing of fluids wherein a rotor is rotated on a shaft and the rotor is provided with an entry for the fluid at center, a pressurizing cavity with vanes, nozzles discharging the fluid forward, a vortex cavity for pressurizing the fluid, and exit means for passing the fluid from the rotor. Usually a casing is provided for collecting the pressurized fluid with an exit for delivery. The pressurizer can be used to pressurize either liquids or gases. The exit means from the rotor may be nozzles directed to discharge the fluid, or they may be other means for reclaiming part of the kinetic energy of the leaving fluid.

3 Claims, 2 Drawing Figures





FLUID PRESSURIZER

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuing-in-part application of "Turbine", filed Aug. 7, 1973, Ser. No. 386,273, now U.S. Pat. No. 3,758,223 and "Reaction Rotor Turbine", filed Sept. 30, 1971, Ser. No. 185,060, now U.S. Pat. No. 3,879,152.

BACKGROUND OF THE INVENTION

This invention relates to fluid pressurizers where centrifugal force is used to increase the pressure of the fluid.

In previous fluid pressurizers of the centrifugal type, fluid is accelerated in the rotor, and then decelerated in the casing producing pressure. These methods are costly in power consumption due to turbulence and friction losses, and for high pressures, the operation may become unstable, which ordinarily is corrected by adding stages, thus increasing the cost of the unit.

SUMMARY OF THE INVENTION

It is an object of this invention to provide a means for pressurizing fluids with a reduced power requirement while still maintaining relatively simple construction for the pressurizer.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of the pressurizer, and FIG. 2 is an end view of the pressurizer with sections removed to show internal details.

Referring to FIG. 1, therein is shown a cross section of the pressurizer along the shaft. 10 is casing, 11 is rotor, 12 is rotor vortex cavity, 13 are vortex cavity exit openings, 14 is fluid exit from casing, 15 are vortex cavity feeder nozzles, 16 is pressurizing cavity, 17 is fluid entry, 18 are vanes, 19 is bearing and seal for shaft 20.

In FIG. 2, 10 is casing, 16 is pressurizing cavity, 18 are vanes, 20 is shaft, 15 are vortex cavity entry nozzles, 12 is vortex cavity, 13 are vortex cavity exit openings, 11 is rotor.

In operation, fluid enters via entry 17 into rotor cavity 16, where it is preliminarily pressurized by centrifugal force with vanes 18 assuring that the fluid will rotate with the rotor 11. The fluid then passes via nozzles 15 forwardly in the direction of rotation into vortex cavity 12, thus adding the fluid exit velocity from the nozzles to the tangential velocity of the nozzles which is due to the rotation of the rotor, and there the fluid is pressurized by centrifugal force due to the fluid being forced to follow a curved path, with higher pressure occurring at the periphery of the said vortex cavity. After pressurization, the fluid is passed from the vortex cavity via openings 13 into casing 10, and from there into exit opening 14. The openings 13 may be plain holes, or they may be orifices arranged to discharge the fluid backward away from the direction of rotation. The rotor is rotated, with power supplied to shaft 20.

The fluid being pressurized may be either a liquid or a gas, or a liquid-gas mixture.

In normal operation, the pressure at the area nearest to the rotor center of cavity 12 is usually zero. This pressure may be higher, as desired. Thus, the fluid velocity within the vortex cavity downstream of nozzles 15 is the sum of fluid velocity leaving said nozzles rela-

tive to rotor, and the rotor velocity. In FIG. 1, two rows of nozzles 15 are shown, more or less rows may be used as desired. Generally, additional nozzle rows will give better control of fluid tangential velocity within vortex cavity 12, thus improving the pressure gain for the pump or compressor.

The shape of the vortex cavity in cross section is shown to be almost triangular. This shape is usually used to control the amount of times that the fluid will have to travel around the vortex cavity. It is generally desirable to limit the number of trips the fluid circulates around the vortex cavity to reduce fluid friction losses. In model studies, it has been desirable to limit these trips to one; that is, the fluid traveling but once around the vortex cavity before discharge from openings 13. The shape of said cavity may be made as desired, to suit the fluid being pressurized, and the amount of pressure gain desired.

The entry pressure of the fluid at the entry 17, may be as desired. As is common with pumps and compressors, there is normally some pressure at entry, such as the pressure of the atmosphere. The performance of the pressurizer of this invention is generally improved with an increase in entry pressure, so that this device is particularly useful as a pressure booster.

The rotor of this pressurizer may be attached to another rotor or two rotors be built together, to form a multistage unit. Such arrangements are particularly desirable where the entry pressure of the fluid entering at 17 is low, while a high fluid exit pressure is required. Obviously, more than two stages may be used, if desired.

The openings 13 may be nozzles arranged to discharge the fluid backward, to reduce work input to rotor. Such is often desirable, as can be readily shown; the following example illustrates this point: Using water as the fluid, a pressure drop of 3 psia produces, when this pressure drop is from nozzle entry to nozzle exit, an exit velocity of 21 FPS, and thus for a small pressure loss at the openings 13 can reduce significantly the work input to the rotor.

In an alternate arrangement, the openings 13 may discharge into an adjacent second rotor mounted concentric with the rotor of the pressurizer, with vanes or buckets being used to convert the kinetic energy of the fluid leaving openings 13, to work. Also, an inward flow type turbine could be used to convert the kinetic energy to work, if desired.

I claim:

1. A fluid pressurizer comprising a shaft mounted for rotation, a rotor mounted on said shaft so as to rotate therewith, a casing enclosing the rotor said rotor having an entry for a fluid to be pressurized near its center, said rotor having a first cavity concentric with said shaft communicating with said entry for passage of said fluid and having a set of vanes, said first cavity having a set of feeder nozzles for discharging said fluid in a direction that is forwardly and in the direction of rotation into a second cavity, said second cavity being a radially outward cavity concentric with said shaft, said fluid being pressurized within said second cavity by centrifugal force acting on said fluid, said fluid being then discharged from said second cavity into the casing through a set of exit openings located near to the outer radius of said second cavity, said fluid pressurizer having an outlet through which the fluid in the casing may flow from the pressurizer.

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2. The pressurizer of claim 1 wherein said set of exit openings are a set of nozzles sized and shaped to accelerate said fluid being discharged from said second cavity.

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3. The pressurizer of claim 1 wherein said set of exit openings are a set of nozzles arranged to discharge said fluid in a direction that is away from the direction of rotation.

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[54] **ROTARY HEAT EXCHANGER WITH COOLING AND REGENERATION**

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee, Houston, Tex. 77042

[22] Filed: **Oct. 1, 1975**

[21] Appl. No.: **618,456**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 474,729, May 30, 1974, and a continuation-in-part of Ser. No. 393,571, Aug. 31, 1973, Pat. No. 3,972,203.

[52] U.S. Cl. **62/401**; 60/682; 62/86; 62/499; 165/88; 165/140; 415/178

[51] Int. Cl.² **F25B 3/00**

[58] Field of Search 415/1, 64, 178; 165/61, 165/88, 140; 416/96; 60/327, 682; 62/86, 401; 122/26; 126/247

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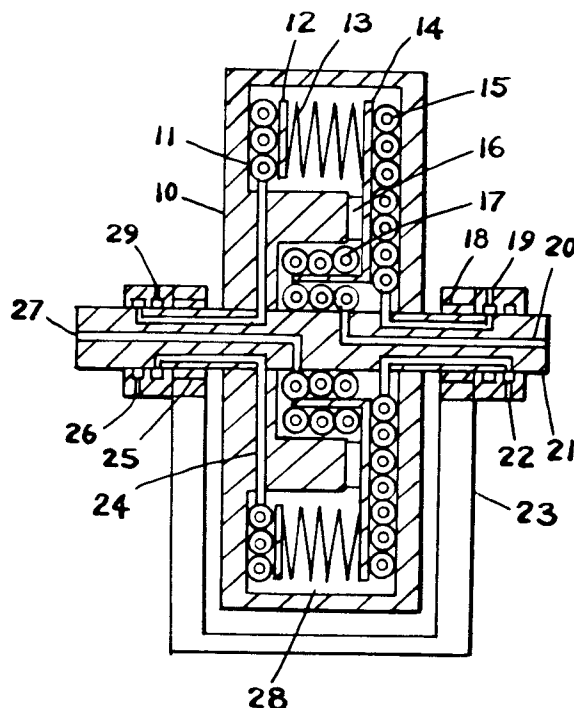
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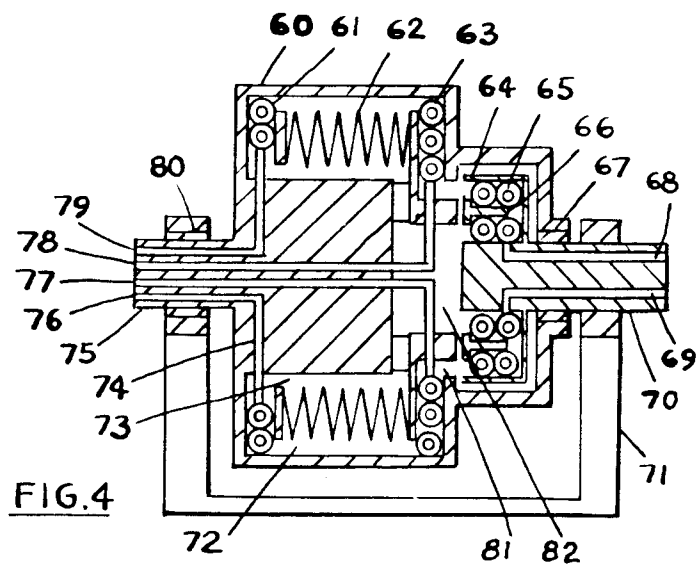
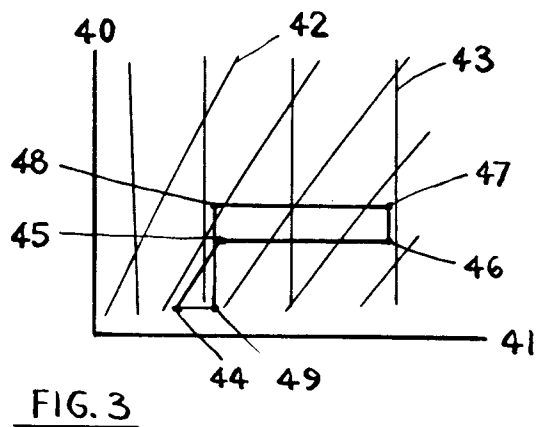
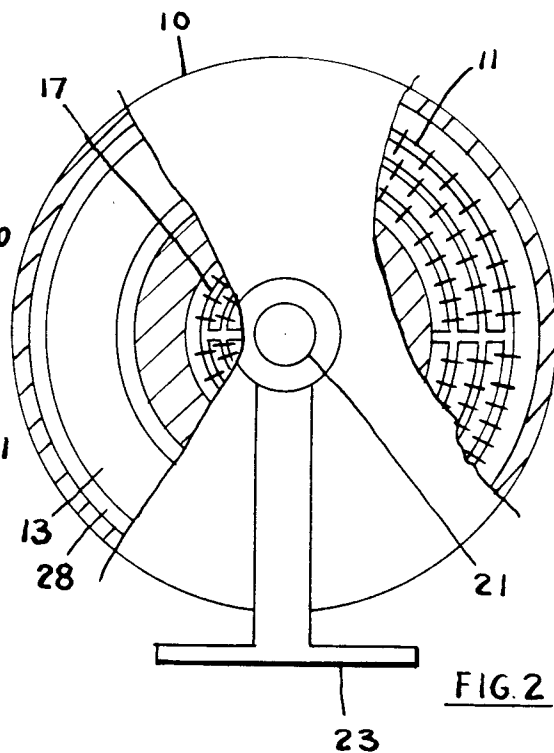
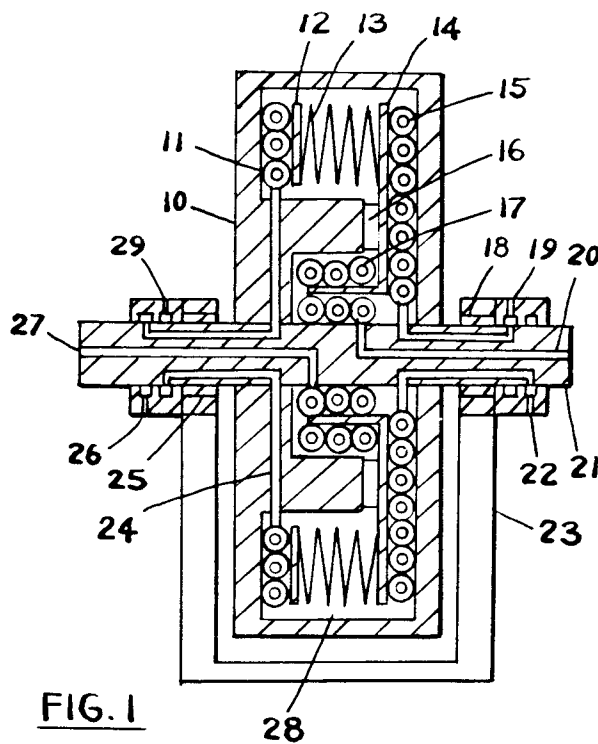
Primary Examiner—C. J. Husar
Assistant Examiner—Sheldon Richter

[57] **ABSTRACT**

A method and apparatus for transport of heat from a low temperature heat source into a higher temperature heated sink, using a compressible working fluid compressed by centrifugal force within a rotating rotor with an accompanying temperature increase. Heat is transferred from the heated working fluid into the heat sink at higher temperature, and heat is added into the working fluid after expansion and cooling from a colder heat source. Cooling is provided within the rotor to control the working fluid density, to assist working fluid circulation. The rotor has outward and inward extending working fluid passages for circulation therein, and heat is provided by a heat source fluid within its own heat exchanger, and heat is delivered into a heated fluid circulating within its own heat exchanger. Cooling is provided by circulating a cooling fluid within its own heat exchanger; alternately, the heated fluid may also serve as the cooling fluid if desired, before being heated. A regenerative heat exchanger is also provided exchanging heat between two streams of the working fluid within the rotor.

6 Claims, 4 Drawing Figures





ROTARY HEAT EXCHANGER WITH COOLING AND REGENERATION

This application is a continuation-in-part application of "Rotary Heat Exchanger with Cooling", filed May 30, 1974, Ser. No. 474,729 and "Rotary Heat Exchanger", filed Aug. 31, 1973, Ser. No. 393,571 now U.S. Pat. No. 3,972,203.

BACKGROUND OF THE INVENTION

This invention relates to devices for producing heating, wherein a fluid releases heat and another fluid gains heat.

The art of generating heat has seen a variety of devices, using the compression of a gas with temperature increase, and subsequent condensing of the fluid, as the means for providing the heat. These devices are inefficient in that they require large amounts of work for their operation.

SUMMARY OF THE INVENTION

It is an object of this invention to provide means for transferring heat from a lower temperature to a higher temperature wherein a working fluid is compressed with a temperature increase within a rotating rotor, with heat supplied into the working fluid from a heat source that is at a lower temperature than the heat delivery temperature, and where a coolant is also used within the rotor for working fluid density control purposes, and wherein the rotor is provided with a regenerative type heat exchanger to exchange heat between two streams of the working fluid within the rotor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial cross section of a rotor, and

FIG. 2 is an end view of the same rotor.

FIG. 3 is a typical pressure-enthalpy diagram for the working fluid circulating within the rotor.

FIG. 4 is an axial cross section of another rotor where one of the heat exchangers used is stationary.

DESCRIPTION OF PREFERRED EMBODIMENTS

The apparatus described in this disclosure is similar to that described in Patent application "Rotary Heat Exchanger with Cooling", filed May 30, 1974, Ser. No. 474,729, except that the regenerative heat exchanger is being added and some other details are being added.

Referring to FIG. 1, therein is shown an axial cross section of a rotor 10, mounted on shaft 21, and supported by bearings 18 and 25, and base 23. The rotor 10 supports heat delivery heat exchanger 11, regenerative heat exchanger 13, heat supply heat exchanger 15 and cooling heat exchanger 17. 12 and 14 are dividers, 16 are vanes, 19 and 22 are heat supply fluid entry and exit, 20 and 27 are cooling fluid entry and exit, 28 is working fluid passage, 24 is heated fluid conduit, 26 and 29 are heated fluid entry and exit.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated. 10 is rotor, 11 is heat delivery heat exchanger, 21 is shaft, 23 is base, 28 is working fluid passage, 13 is regenerative heat exchanger, and 17 is cooling heat exchanger.

In FIG. 3, a pressure-enthalpy diagram for a typical working fluid is illustrated. 40 is pressure line and 41 is enthalpy line. 42 is constant entropy line and 43 is constant temperature line. The fluid is compressed from 44 to 45 and then heated from 45 to 46 by heat

from the same fluid which is cooled from 47 and 48; with compression and heat removal from 46 to 47. Expansion of the fluid is from 48 to 49 with heat addition, and the fluid is then cooled from 49 to 44. Heat is delivered from 46 to 47, and heat is received from 48 to 49.

In FIG. 4, another cross section of another rotor is shown, where the cooling heat exchanger is kept stationary. 60 is rotor, 61 is heat delivery heat exchanger, 62 is regenerative heat exchanger, 63 is heat supply heat exchanger, 64 is baffle and part of stationary heat exchanger 65, 66 is another baffle, 67 is bearing and seal, 68 is coolant passage, 69 is another coolant passage, 70 is stationary shaft, 71 is base, 72 is working fluid passage, 73 is working fluid passage, 74 is heated fluid conduit, 75 is rotor shaft, 76 and 79 are heated fluid passages, 77 and 78 are heat source fluid passages, and 80 is bearing. 81 and 82 are working fluid passages.

In operation, the rotor is caused to rotate, and is filled with a suitable working fluid. Heated fluid and heat source fluid are circulated from external sources. The working fluid is compressed by centrifugal force in the rotor internal outward extending passages and during and after such compression heat is removed from the working fluid in heat exchanger 11 into the fluid to be heated. After this, the working fluid passes along passages 28 and delivers heat into another stream of the working fluid in heat exchanger 13, and then the working fluid is expanded with temperature decrease, and during and after such expansion heat is added into the working fluid from a heat source fluid in heat exchanger 15. After this, the working fluid is cooled in heat exchanger 17 to increase the working fluid density to obtain working fluid circulation within the rotor; this density increases the weight of the working fluid within the compression side of the rotor assist or to provide for circulation. Thus, heat is provided into the working fluid by heat source fluid, and heat is delivered by the working fluid at a higher temperature into the heated fluid. The regenerator heat exchanger makes it possible to maintain relatively moderate rotor speeds while still having a relatively high temperature in the heat delivery heat exchanger.

The regenerative heat exchanger was described and shown in several copending patent applications: "Turbine", filed Apr. 9, 1975, Ser. No. 566,373; and "Heat Pump", filed Dec. 18, 1974, Ser. No. 533,983. The various forms of the heat exchanger shown in the previous patent applications for the regenerative heat exchanger can be also used with the heat pump with cooling of this invention. Similarly, the heat exchanger of FIG. 3, of patent application "Heat Exchanger", filed July 14, 1975, Ser. No. 595,389, can be used with the unit of this invention.

The heat exchangers 11 and 15 and 17 are shown to be made of finned tubing with a fluid circulated within the tubes. Other types of heat exchangers may be used, as desired, especially if the fluids circulated within the heat exchangers are gaseous.

The working fluid is normally a gas, and the heated fluid and the heat source fluid and the coolant may be either gases or liquids.

Vanes may be employed in the various working fluid passages as required to prevent tangential movement of the working fluid. Alternatively, such vanes may be curved or their radial lengths adjusted to provide for controlled tangential movement for the working fluid.

such adjustments may be used to assist in the circulation of the working fluid.

In the rotor of the apparatus disclosed herein, the working fluid circulation is assured mainly or solely by density control of the working fluid within rotor passages, as indicated in FIG. 3. Where the temperature of available heat source is not suited for obtaining circulation by density control alone, work may be added into the working fluid from the rotor by manipulating the curvature or radial length of the radial working fluid passages within the rotor. Where the temperature of the heat source fluid is sufficient, and other temperatures suitable, then the working fluid is circulated by density control alone, and the working fluid will rotate with the rotor in all areas of the rotor, and also, for such conditions, the work input to the rotor shaft is that which is required to rotate the rotor against friction losses. Also, during such operating conditions, part of the heat supplied by the heat source fluid is passed into the heated fluid, and a part is passed into the coolant.

The use of the regenerative heat exchanger allows the rotor speed to be reduced, while providing a means for high heat delivery temperatures. However, the rotor can be rotated at high speeds, and for such applications, the rotor is usually encased within a vacuum jacket to reduce and eliminate fluid friction losses on rotor external surfaces.

Various regulators and controls are used with the device of this invention. They do not form a part of this invention and are not further described herein.

In FIG. 3, it is shown that the working fluid pressure is constant, normally, from 49 to 44, and in FIG. 4, the cooling heat exchanger portion is stationary with a constant pressure therein. Thus, it should be obvious, that the cooling heat exchanger could be deleted, and the working fluid passed from the rotor to space outside rotor, if desired, with the working fluid leaving the rotor downstream of heat exchanger 63, via opening 81, and then entering the rotor via opening 82. In this manner, the heat exchanger 65 is eliminated, with cooling done outside rotor. Alternately, ambient air then could be used as the working fluid. Similar rearranging could be also carried out for the unit shown in FIG. 1. Further, backward oriented nozzles can be installed in opening 81, to recover work from departing working fluid, as shown in my co-pending patent application "Heat Exchanger", Ser. No. 423,560, abandoned filed

Dec. 10, 1973. In deleting the heat exchanger 65, items 64, 66, 67, 68, 69 and 70 are also deleted, as well as the rotor extension surrounding heat exchanger 65.

I claim:

1. In a rotary heat exchanger wherein a compressible working fluid is circulated within a rotor outwardly in a first passage and inwardly toward center of rotation in a second passage, with the inward ends and outward ends of said passages connected by passage means to allow the circulation of said working fluid, and wherein a heat delivery heat exchanger is provided to remove heat from the working fluid during and after compression, and a heat source heat exchanger is provided to add heat into the working fluid during and after expansion, the improvement comprising:

a. a regenerative heat exchanger provided to exchange heat between two streams of the working fluid, with the working fluid stream leaving said heat delivery heat exchanger being in heat exchange relationship with the working fluid stream entering said heat delivery heat exchanger and being upstream of said heat delivery heat exchanger, and wherein a cooling means is provided to remove heat from said working fluid downstream of said heat source heat exchanger.

2. The rotary heat exchanger of claim 1 wherein said heat delivery heat exchanger is provided with a heated fluid circulated within said heat delivery heat exchanger.

3. The rotary heat exchanger of claim 1 wherein said heat source heat exchanger is provided with a heat source fluid circulated within said heat source heat exchanger.

4. The rotary heat exchanger of claim 1 wherein said cooling means comprises a stationary cooling heat exchanger for removal of heat from said working fluid, into a coolant circulated within said cooling heat exchanger.

5. The rotary heat exchanger of claim 1 wherein said cooling means is a cooling heat exchanger carried by said rotor for removal of heat from said working fluid.

6. The rotary heat exchanger of claim 1 wherein said heat source heat exchanger is provided with a heat source fluid circulated within said heat source heat exchanger and wherein said heat source fluid is a gas at least when entering said heat source heat exchanger entry passages.

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[54] ROTOR WITH RECIRCULATION

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[22] Filed: Nov. 28, 1975

[21] Appl. No.: 636,310

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 501,064, Aug. 27, 1974, Pat. No. 3,939,661.

[52] U.S. Cl. 415/53 R; 415/80

[51] Int. Cl.² F04D 5/00

[58] Field of Search 415/80, 52, 53; 60/482; 416/179

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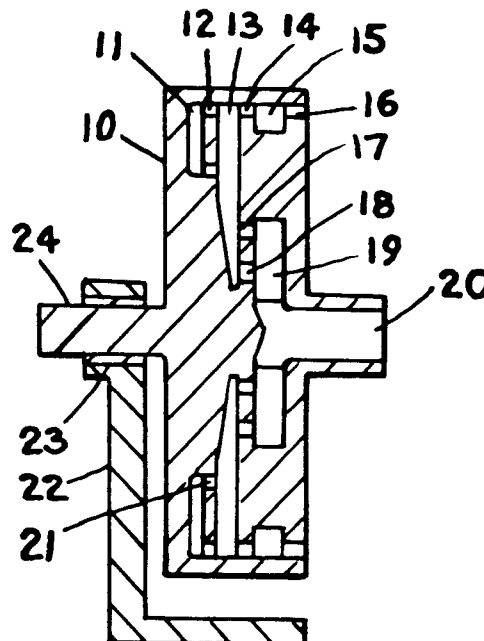
Primary Examiner—Henry F. Raduazo

[57]

ABSTRACT

A method and apparatus for generating power and for pumping fluids, wherein a free vortex cavity, within a rotating rotor, is used to pressurize a fluid after which the fluid may be used as a pressurized fluid or be used to generate power. The working fluid is injected into the free vortex cavity through feed nozzles oriented to discharge the working fluid forwardly in the direction of rotation, so that the working fluid rotational speed is normally, at least in part of the cavity, greater than the rotational speed of the rotor. The working fluid is pressurized within the vortex cavity by being forced to follow a curved path. Part of the working fluid is taken near the periphery of the curved passage, and recirculated through nozzles located toward the rotor center from the periphery thus providing additional fluid flow within the free vortex cavity and improving the rotor performance. The recirculation passages may be either radial, or be curved, and the recirculation nozzles are arranged to discharge the recirculation fluid forwardly. Fluids used as the working fluid may be liquids or gases. The rotor is provided with a shaft, and is rotatably mounted.

5 Claims, 5 Drawing Figures



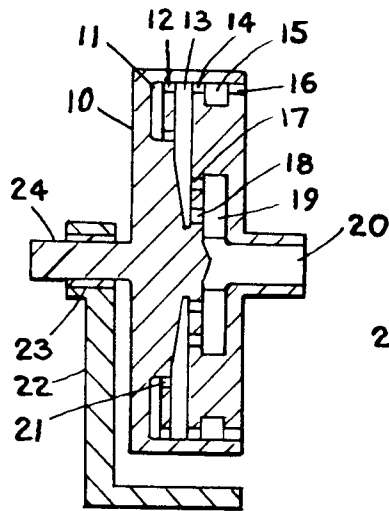


FIG. 1

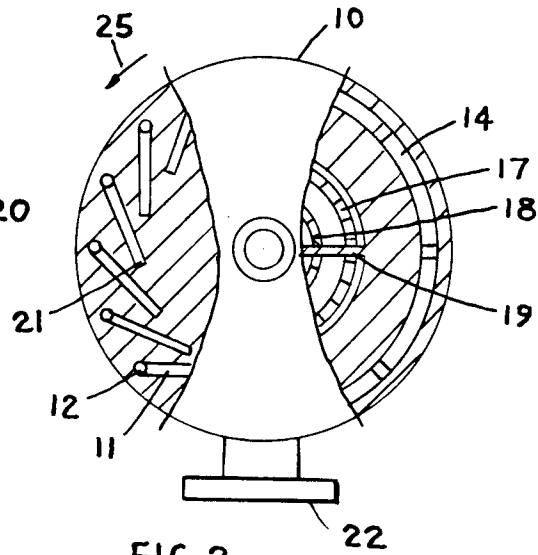


FIG. 2

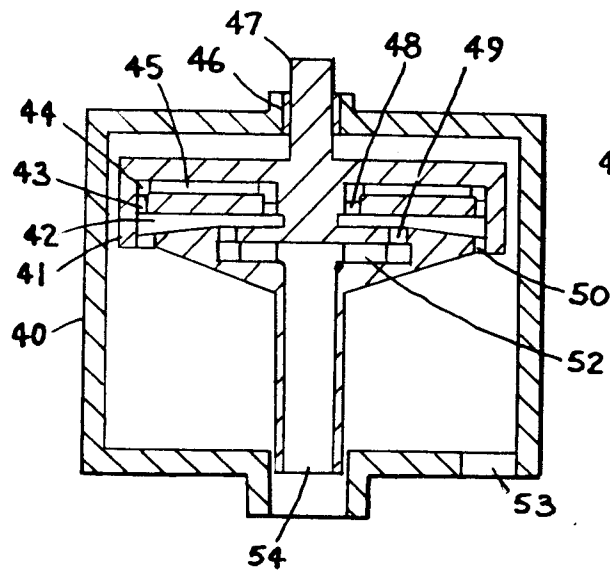


FIG. 3

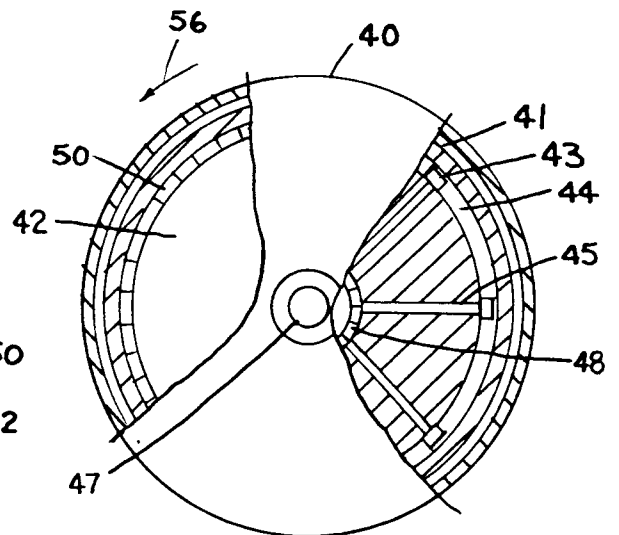


FIG. 4

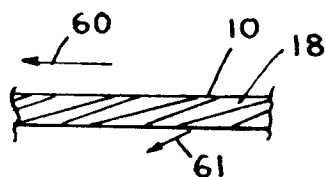


FIG. 5

ROTOR WITH RECIRCULATION **CROSS REFERENCES TO RELATED** **APPLICATIONS**

This application is a continuation-in-part application of "Power Generator," filed 8-27-74, Ser. No. 501,064 now U.S. Pat. No. 3,939,661.

BACKGROUND AND SUMMARY OF THE **INVENTION**

This invention is concerned with turbines and fluid pressurizers, where a free vortex is used within a rotating rotor to pressurize a working fluid.

In my previous U.S. Pat. No. 3,879,152, "TURBINE," and in U.S. Pat. No. 3,758,223, "REACTION ROTOR TURBINE," I had described turbines where a free vortex cavity within a rotating rotor is used to create a free vortex where a working fluid is pressurized by centrifugal action. These turbines have relatively high friction losses within the rotor free vortex cavity due to the necessity of using relatively small cavities in many instances, with accompanying large velocity differentials between the working fluid and the rotor wall confining the working fluid. There are various ways to help to reduce the friction loss, and one such is to increase the fluid flow. To obtain the necessary increased fluid flow, within the free vortex cavity, recirculation may be used.

It is an object of this invention to provide a method and apparatus, to pressurize fluids within a rotor cavity, and to generate power, where a portion of the working fluid flowing through said cavity is recirculated using passages built into the rotor to pass the fluid from the area near rotor periphery into forwardly discharging nozzles located inward toward rotor center from the rotor periphery. It is also the object of this invention to provide a pumping means to pressurize fluids.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the turbine, and

FIG. 2 is an end view of the unit shown in FIG. 1,

FIG. 3 is a cross section of another arrangement of the unit and FIG. 4 is an end view of the unit of FIG. 3,

FIG. 5 is a nozzle detail.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, therein is shown an axial cross section of the unit. Rotor 10 is rotatably supported by shaft 24, bearing 23 and support 22. Working fluid enters via entry port 20, and passes to spaces defined by vanes 19, and from there via nozzles 18 and 17 to free vortex cavity 13, with nozzles 17 and 18 being oriented to discharge the working fluid forwardly. After compression, part of the working fluid is recirculated via passages 12 and 11 into nozzles 21 to be injected into said cavity forwardly. Also, after compression, a part of the working fluid is passed from the vortex cavity via openings 14 to annular spaces 15, and from there discharged via passages 16, which may be nozzles oriented to discharge backward the working fluid thus generating torque on the rotor.

In FIG. 2, an end view of the unit of FIG. 1 is shown. 10 is rotor, 14 is fluid passage, 17 are nozzles, 18 are nozzles, 19 are vanes which may be radial or curved, 22 is base, 11 and 12 are fluid passages, shown here pass-

ing the fluid forward into nozzles 21, and 25 indicates direction of rotation for the rotor.

In FIG. 3, another arrangement of the unit is shown, with the unit mounted within a closure for collection of the fluid leaving the rotor, and also for allowing for the pressurization of the casing if desired. 40 is casing supporting rotor bearing 46 and rotor shaft 47 and having fluid entry 54 and exit 53. Rotor is 41, having fluid passages defined by vanes 52, and nozzles 49 for feeding the working fluid from entry into rotor vortex cavity 42. After compression, part of the fluid is passed via passages 43, 44 and 45 into injection nozzles 48, and the remainder of the working fluid is passed via exit openings 50 which may be nozzles, into casing 40 and from there to discharge 53.

In FIG. 4, an end view of the unit shown in FIG. 3 is illustrated. 40 is casing, 41 is rotor, 43, 44 and 45 are recirculation fluid passages, 48 are recirculation nozzles, 47 is shaft, 42 is vortex cavity, 50 are exit openings, and 56 indicates a direction of rotation for the rotor.

In FIG. 5, a detail of nozzles is shown, with 60 indicating of movement, 61 of fluid leaving nozzles, 10 being rotor and 11 being the nozzles.

The operation is as indicated hereinbefore. When the unit is used as a turbine, a working fluid is passed via entry into the rotor where the working fluid is pressurized first in the entry defined by vanes 19, and then further pressurized in the vortex cavity, after which the fluid leaves via exits 16 oriented to discharge backward to generate torque on the rotor and thus generate power. Part of the fluid is circulated within the rotor as indicated to provide for improved performance.

When the unit is used as fluid pressurizer, the exit pressure from the unit is maintained at desired value, and the exit nozzles or openings 50 are made for reduced pressure drop as may be desirable.

The free vortex cavity is usually made in tapered form as shown in the drawings, to provide for desired velocity pattern within the cavity. The nozzles for recirculation may be placed as desired, and as indicated in the drawings; one or more nozzle rows may be used. Similarly, one or more rows of feeder nozzles may be used.

The working fluid recirculation passages may be radial open channels, or they may be curved in various ways, depending of the amount of pressure available within the vortex cavity.

The drawings show only a single stage for the turbine and compressor. These units may be built in multiple stages if desired, with the discharge opening, such as 50, connecting with entry side passages defined by vanes 52. By using multiple stages, larger pressure drops can be accommodated in a single rotor, and when used as a pressurizer, large delivery fluid pressures can be obtained.

Applications include power generators for liquid and gaseous fluids, and as pumps for liquids, and gas compressors. Working fluid may be either a liquid, a gas, or a liquid-gas mixture.

The working fluid is shown to be discharged from the rotor via exit nozzles, such as item 16, FIG. 1. These nozzles are shown discharging the working fluid axially away from the rotor, and tangentially backward to generate thrust, and torque on the rotor. These nozzle may be also located at the periphery to discharge radially and backward to generate said torque, and such arrangement is particularly suited for pump applica-

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tions where a standard pump volute casing may be used to surround the rotor to receive the fluid being discharged by the rotor, from the peripheral nozzles.

Other types exit means for the working fluid may be also used, in place of the nozzle 16. Such means may include reaction vanes, a second rotor to generate work from the pressurized fluid being discharged from cavity 13, and other means. In my co-pending patent application "Power Generator," a second rotor is being used. Said application is Ser. No. 501,064, and was filed 8-27-74.

The fluid passages for recirculation may be also used to add additional pressure into the fluid being recirculated; this is shown in FIG. 2, passages 11, where the passages are oriented to pass the fluid in the forward direction as the rotor rotates, so that the exit opening from the passage leads the entry opening. The pressure increase within the passage depends on the placement of these passages, and their specific curvature. In this manner, one may select the pressure that is desired at exit openings 21. It should be noted, that if the passages 11 are made such that the exit opening 21 lags behind the entry 12, then there will be a pressure decrease within the passage 11, during the rotation of the rotor.

I claim:

1. A rotor for pressurizing of a fluid and comprising:
 - a. a fluid entry to said rotor near the center of rotation of said rotor;
 - b. a set of fluid nozzles for passing said fluid and oriented to discharge said fluid forwardly in the direction of rotation;
 - c. a vortex cavity within said rotor to receive said fluid from said fluid nozzles; for pressurizing said fluid;

- d. a fluid discharge means from said vortex cavity adapted for discharging the fluid that entered the rotor through said fluid entry;
- e. a fluid recirculation means for recirculating a part of said fluid from said vortex cavity into fluid recirculation nozzles; with the fluid recirculation nozzles discharging said fluid forwardly into said vortex cavity in the direction of rotation; with the recirculation nozzles being located inwardly toward rotor center from the vortex cavity area where the fluid is taken from said cavity for recirculation.

2. The rotor of claim 1 wherein said fluid discharge means are a set of fluid discharge nozzles arranged to discharge said fluid backward away from the direction of rotation.

3. The rotor of claim 1 wherein the fluid passages for recirculation fluid receive their fluid at rotor periphery.

4. In a rotor wherein a fluid is pressurized in a forced vortex confined within rotor passages, then is discharged via nozzles forwardly into a free vortex cavity within said rotor to be pressurized, and then is discharged from said rotor via exit openings, the improvement comprising:

- a. a means for recirculating a portion of said fluid that is being circulated within said free vortex cavity by providing a passage for said fluid to pass a portion of the fluid circulating within said rotor into a recirculation opening arranged to discharge said fluid forwardly in the direction of rotation into said free vortex cavity, with the entry for said fluid into said passage being further away from the rotor center than said recirculation opening discharging into said free vortex cavity.

5. The rotor of claim 4 wherein said passage is arranged to pass said fluid forwardly in the direction of rotation with said recirculation opening leading the entry opening during rotation of the rotor.

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[54] ROTOR WITH JET NOZZLES

3,945,757 3/1976 Cummins 415/80

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Houston, Tex. 77042

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[21] Appl. No.: 659,797

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Related U.S. Application Data

Primary Examiner—C. J. Husar

[63] Continuation-in-part of Ser. No. 568,895, April 17,
1975, Pat. No. 4,003,673.

[52] U.S. Cl. 415/80; 415/1

[51] Int. Cl.² F01D 1/18

[58] Field of Search 415/80, 109, 1;
60/39.35

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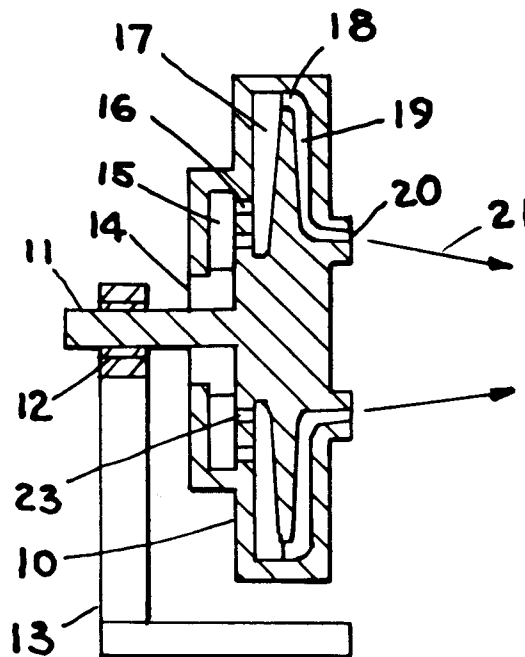
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3,879,152	4/1975	Eskeli	415/80

[57] ABSTRACT

A method and apparatus for the generation of high velocity fluid jets by using a rotor with attached discharge passages and nozzles. A working fluid is accelerated by using a rotating rotor, either of the free vortex type or of the forced vortex type, and the accelerated fluid is then passed through forwardly and inwardly extending passages, and discharged at high speed through openings that are inward from the rotor periphery. Fluids may be either liquids or gases. The high velocity fluid jet may be used where such jets are required, such as excavation, cutting, or as a source of thrust. The rotors may be also made multistage, if additional jet velocity is required.

5 Claims, 6 Drawing Figures



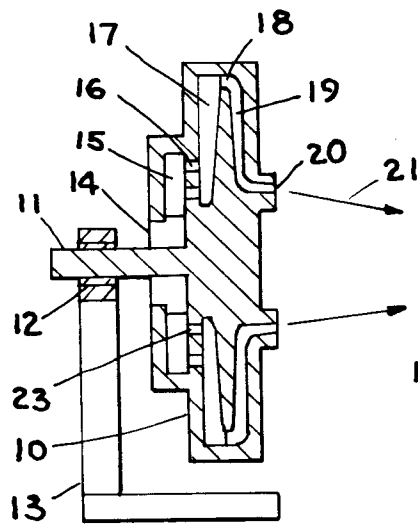


FIG. 1

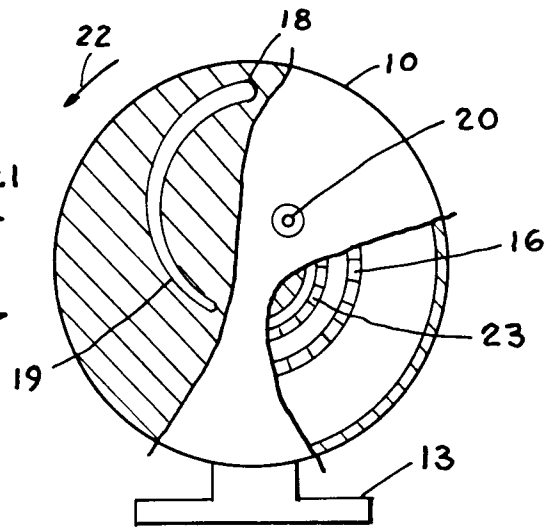


FIG. 2

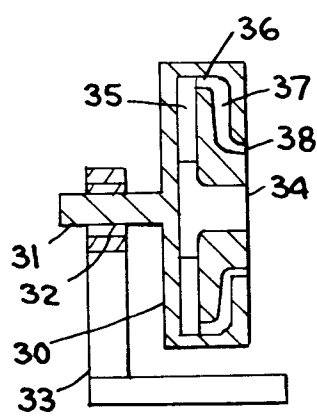


FIG. 3

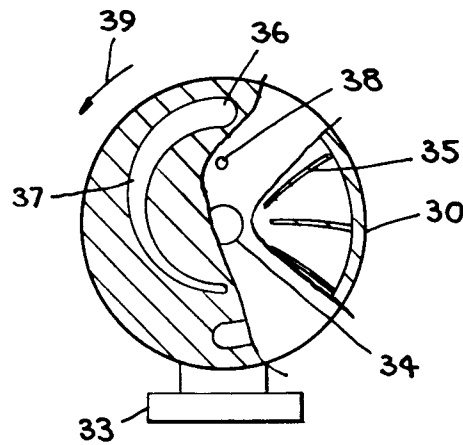


FIG. 4

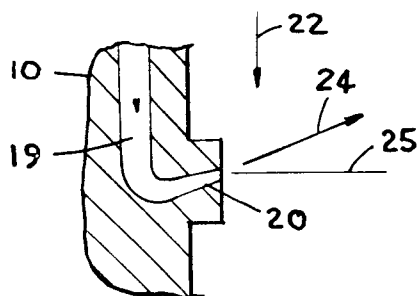


FIG. 5

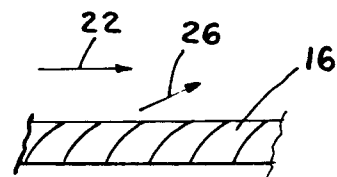


FIG. 6

ROTOR WITH JET NOZZLES

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part application of "Fluid Pressurizer", Ser. No. 568,895, filed 4/17/75, now U.S. Pat. No. 4,003,673.

BACKGROUND OF THE INVENTION

This invention relates generally to apparatus for providing a high velocity fluid jet.

Previously, high velocity fluid jets have been provided by using a pump to force a fluid at high pressure through a nozzle, thus providing for fluid acceleration. Such apparatus is costly due to the heavy equipment required, and uses large amounts of power.

SUMMARY OF THE INVENTION

The object of this invention is to provide a relatively simple means for generating a high velocity fluid jet, in a single rotating rotor, using the rotor to accelerate a working fluid, and then passing the working fluid via passages that allow the fluid to travel in the tangential direction faster than the local rotor speed, with the fluid passages leading the fluid inward toward the rotor center. The fluid is then discharged from the rotor through openings or nozzles at a high velocity, usually at an absolute direction that is nearly parallel to the rotor shaft.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the apparatus, and

FIG. 2 is an end view of the unit of FIG. 1.

FIG. 3 is a cross section of another form of the apparatus, and FIG. 4 is an end view of the unit of FIG. 3.

FIG. 5 is a discharge nozzle detail.

FIG. 6 is a detail of one form of the feeder nozzles used in the unit of FIG. 1.

DESCRIPTION OF PREFERRED EMBODIMENTS

In FIG. 1, the rotor 10 is supported by shaft 11, bearings 12 and base 13. Working fluid enters the rotor via opening 14 and is first pressurized in passages defined by vanes 15, and then passed through forwardly oriented nozzle rows 16 and 23 into free vortex cavity 17, with pressurization there, the fluid passes via openings 18 into passages 19, and from there via exit openings 20 leaves the rotor. 21 indicates that the jets leaving openings 20 may be arranged to converge at a point some distance away.

In FIG. 2, 10 is rotor, 20 is exit, 16 and 23 are feeder nozzle rows, 13 is base, 19 is working fluid passage, and 22 indicates direction of rotation of rotor.

In FIG. 3, a cross section of another form of the apparatus is shown. Rotor 30 is supported by shaft 31, bearing 32 and base 33. Fluid enters via opening 34, passes via radial passages defined by vanes 35, and passes via openings 36 into passages 37, and from there to exit openings 38.

In FIG. 4, an end view of unit of FIG. 3 is illustrated. 30 is rotor, 34 is fluid entry, 33 is base, 37 is fluid passage, 39 indicates direction of rotation for rotor, 36 is fluid passage, 38 is fluid exit, and 35 is vane in outward extending passage.

In FIG. 5, a detail of the rotor exit nozzles or openings is shown. The section of this detail is 90° transverse

to the section of FIG. 1. 10 is rotor, 22 indicates rotor rotation, 19 is fluid passage, 20 is exit opening or nozzle, 24 indicates direction of fluid leaving the opening, with the leaving direction usually being backward with respect to the rotor rotation direction, and 25 indicates orientation of rotor shaft.

In FIG. 6, a detail of the feeder nozzles as used in the unit of FIG. 1, is illustrated. 22 indicates the direction of nozzle movement, 16 are the nozzles, and 26 indicates the direction of leaving fluid, with the direction of the leaving fluid being forwardly.

In operation, in the unit shown in FIG. 1, the working fluid is pressurized in the forced vortex cavity formed by vanes 15, and then discharged via the feeder nozzles into the free vortex cavity 17 in forwardly direction so that the tangential velocity of the rotor, and the leaving velocity of the fluid are added. In the free vortex cavity 17 the fluid is pressurized further and then the pressurized fluid is passed into discharge passages 19, which are oriented to allow the fluid to travel at a higher tangential velocity than the rotor. After passage through passage 19, the fluid is passed out from the rotor via openings 20. Rotor is rotated by a drive connected to shaft.

The operation of the unit of FIG. 3 is similar, except that the free vortex cavity is omitted, and the rotor has a forced vortex only to pressurize and accelerate the working fluid.

The passages 19 and 37 are shown in the drawings to be curved and tapering. The curving provides for best efficiency, but is not mandatory; straight forward passages, with their exit nearer to rotor center than the entry, can be used also. The tapering of passages 19 and 37 also is for best efficiency, to provide for cross sectional area of the passage to match the fluid velocity.

The exit openings 20 and 38 are usually nozzles to provide for additional acceleration of the fluid, and to utilize any available pressure that may still be remaining in the fluid. These nozzles are usually oriented as shown in FIG. 5, to provide for axial discharge of the working fluid from the rotor. Further, the nozzles may be arranged to converge the fluid streams into a single point as shown in FIG. 1, external to the rotor. Obviously, the nozzles 20 or nozzles 38, may be also brought into a single point at an area near rotor center, to discharge the working fluid through a single opening or nozzle. A single exit opening near rotor center may be desirable in instances where the working fluid pressure is sufficient at rotor periphery, and a single fluid jet is desired. In FIG. 1 and FIG. 2, two passages 19 are shown; one passage 19 may be used, or more than one may be used as desired.

The unit of FIG. 1 may be made multistage, by using the fluid leaving via openings 20 as the feed fluid for the next stage free vortex cavity; in such arrangement, nozzles 20 will replace nozzles 16 and 23. Alternately, the device of FIG. 3, may be used as the first stage, and the second stage may be similar to the unit of FIG. 1. As is obvious, various combinations of the units of FIG. 1 and FIG. 3 may be used.

Also, the unit of FIG. 1 may be made into a regenerative unit by feeding a part of the fluid passing through passage 19 into free vortex cavity 17, with the feeder arranged to discharge the recirculated fluid forwardly to assist in the circulation of the working fluid within the free vortex cavity. Such arrangement is shown in a

co-pending patent application. "Rotor with Recirculation", filed Nov. 28, 1975, Ser. No. 636,310.

By using multistage arrangements, combinations of the unit of FIG. 1 and FIG. 3, and recirculation, the rotor speeds can be reduced, thus making the unit lighter in weight, and less costly.

Applications for this apparatus include cutting of materials, use in excavation of rock and soil, in thrust generation and in all uses where a high velocity fluid jet is desired.

Working fluids may be either liquids, or gases.

The rotor of the form of FIG. 1 was the subject of my previous U.S. Pat. No. 3,879,152.

I claim:

1. In a rotating rotor, wherein a working fluid enters said rotor through an entry and is accelerated by said rotor, the improvement comprising:

- a. discharging said working fluid through an inwardly extending fluid passage, with said passage being arranged to allow movement of said working fluid forwardly in the direction of rotation for additional acceleration of said working fluid, with said working fluid being discharged from said rotor via an

exit opening communicating with the inward end of said fluid passage.

2. The rotor of claim 1 wherein said fluid passage is curved forwardly in the direction of rotation to pass said working fluid spirally inwardly to allow acceleration of said working fluid within said fluid passage before leaving said rotor.

3. The rotor of claim 1 wherein said exit opening is a converging nozzle.

4. A method of generating a high velocity fluid jet comprising the steps of:

- a. accelerating a fluid in a rotating rotor to a tangential velocity that is approximately the same as the tangential velocity of said rotor in the area near rotor periphery;
- b. further accelerating said fluid within said rotor in an inwardly and forwardly extending fluid passage;
- c. discharging said fluid from said rotor via an exit opening that is nearer to the rotor center than the rotor periphery.

5. The method of claim 4 comprising the further step of accelerating said fluid further in exit nozzles that are said exit openings.

* * * * *

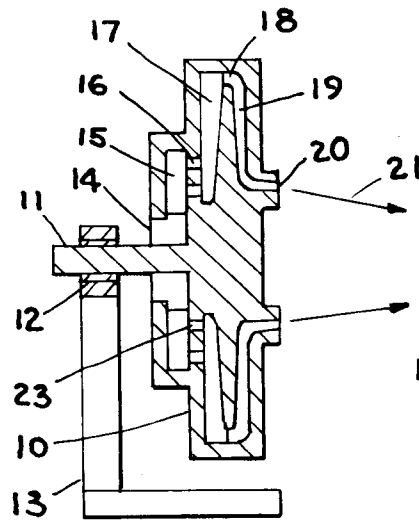


FIG. 1

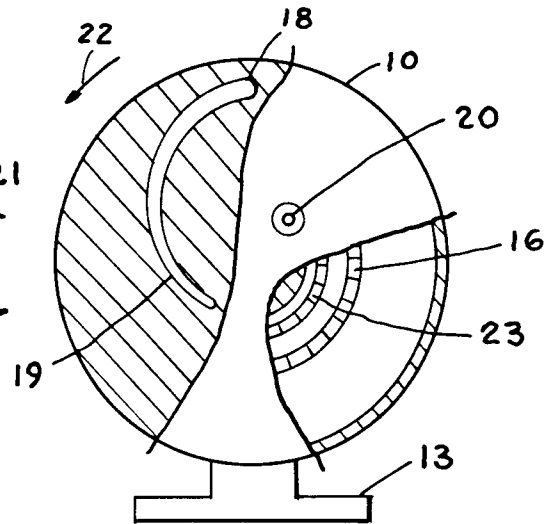


FIG. 2

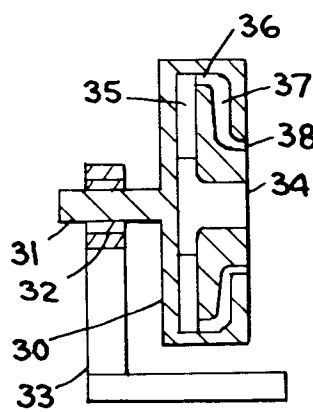


FIG. 3

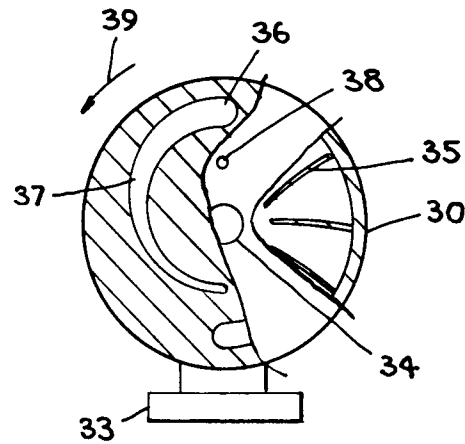


FIG. 4

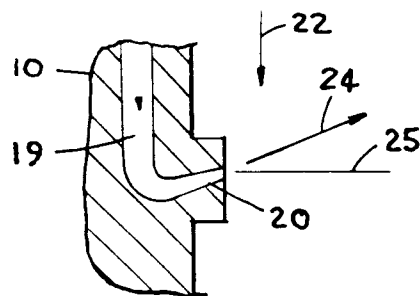


FIG. 5

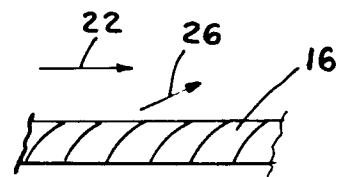


FIG. 6

[54] HEAT EXCHANGER

3,949,557 4/1976 Eskeli 60/682

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee,
Houston, Tex. 77042

FOREIGN PATENT DOCUMENTS

26,184 11/1907 United Kingdom 165/86

[21] Appl. No.: 595,389

[22] Filed: July 14, 1975

Primary Examiner—Ronald H. Lazarus
Assistant Examiner—Sheldon Richter

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 537,067, Dec. 30,
1974, Pat. No. 3,931,713, and Ser. No. 591,881, June 30,
1975, abandoned.

[51] Int. Cl.² F25B 3/00

[52] U.S. Cl. 165/88; 60/682;
62/86; 62/401; 62/499

[58] Field of Search 62/401, 403, 499, 86,
62/87, 402; 165/86, 88; 60/682

[56] References Cited

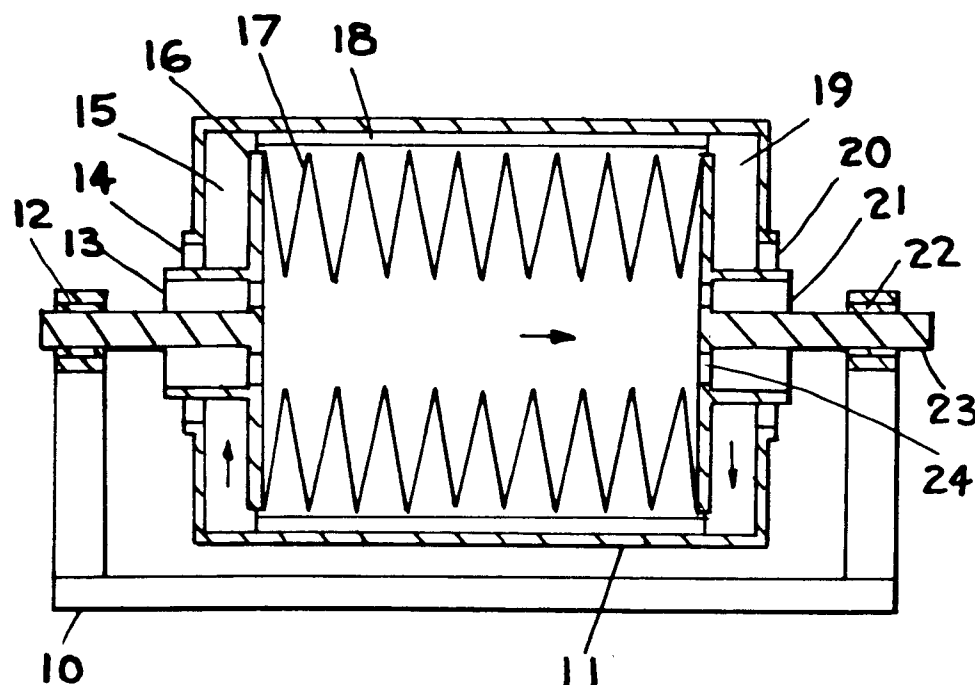
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3,791,167	2/1974	Eskeli	62/401

[57] ABSTRACT

A method and apparatus for exchanging heat between two fluid streams, wherein the two fluids are passed in heat exchange relationship with each other within a rotating heat exchanger mounted on a shaft. The heat exchanger comprises a rotating housing, with the heat transfer elements mounted within, so that one of the fluid streams is passed through areas near the housing periphery, and the other fluid stream is passed through areas that are inward nearer to the axis of rotation from the housing periphery. The heat exchanger elements are made of heat conductive material and form bellows like pockets for the heat exchange to take place. Various vanes or other means may be provided to reduce tangential movement of the fluids, and thermal insulation layers may also be provided to prevent undesirable heat transfer. Fluids passing through the heat exchanger may be either gases or liquids.

5 Claims, 3 Drawing Figures



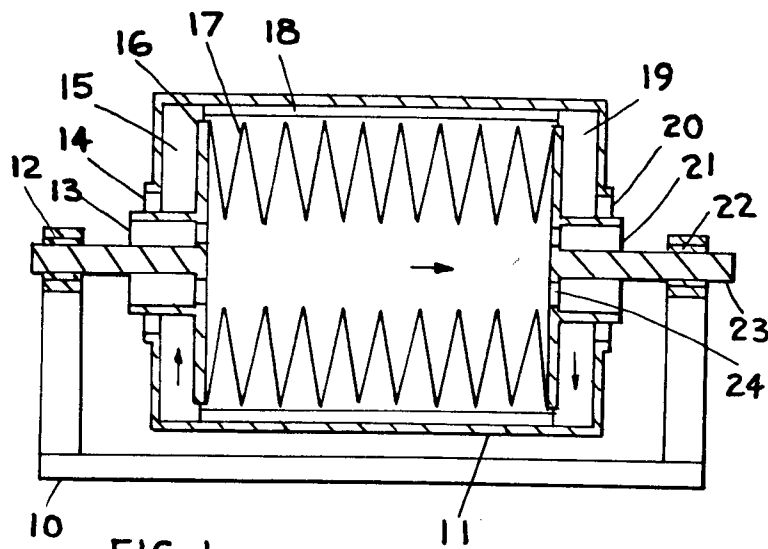


FIG. 1

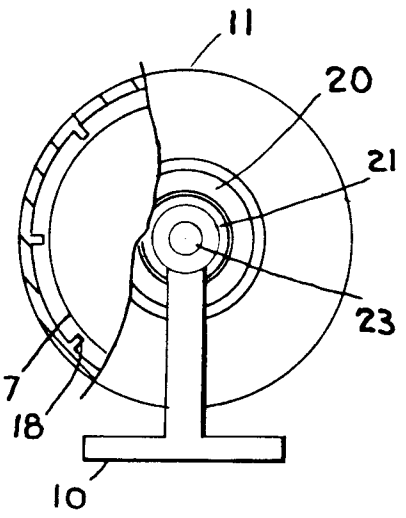


FIG. 2

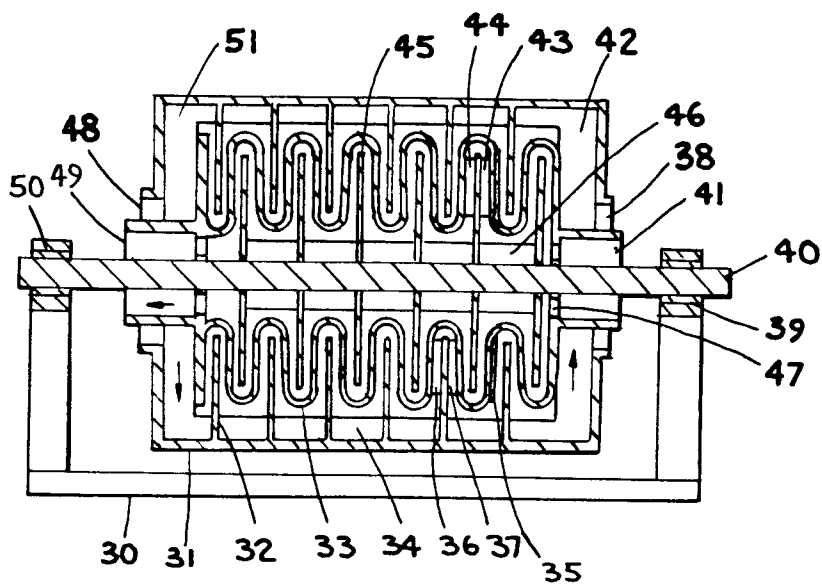


FIG. 3

HEAT EXCHANGER

CROSS REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part application of "Turbine with Regenerator" filed Dec. 30, 1974 Ser. No. 537,067, now U.S. Pat. No. 3,931,713, and "Heat Pump", Filed on or about June 30, 1975, Ser. No. 591,881 now abandoned.

BACKGROUND OF THE INVENTION

This invention relates to heat exchangers of the type for exchanging heat primarily between two streams of fluids that are gaseous, and wherein the heat exchanger elements rotate to enhance heat transfer.

There have been various types of gas-to-gas heat exchangers in the past, where large surface areas have been provided for both fluids to effect necessary heat transfer. These heat exchangers are costly, due to the low heat transfer coefficients that gaseous fluids have, especially where the gas density is relatively low; also, such heat exchangers require a large volume to contain the necessary surface areas.

SUMMARY OF THE INVENTION:

It is an object of this invention to provide a heat exchanger wherein centrifugal force is employed to improve heat transfer between fluid streams, and particularly between two gaseous fluid streams. It is also an object to provide a heat exchanger wherein a large surface area with high thermal conductivity is contained within a relatively small volume thus reducing the cost of such heat exchanger. Also, by use of thermal insulation in suitable areas of the heat transfer elements, it is possible to improve the flow characteristics of the heat exchanger for the fluids passing through.

BRIEF DESCRIPTION OF THE DRAWINGS:

FIG. 1 is a cross section of a form of the heat exchanger, and

FIG. 2 is an end view of the unit shown in FIG. 1.

FIG. 3 is a cross section of a unit that is a modified form of the unit shown in FIG. 1.

DESCRIPTION OF PREFERRED EMBODIMENTS:

Referring to FIG. 1, therein is shown an axial cross section of the heat exchanger. 10 is base supporting bearings 12 and 22, and shaft 23. 11 is heat exchanger housing, which ordinarily is a circular cylinder, and first fluid stream enters via opening 20, passes between vanes 19, 18 and 15, and is discharged via opening 14. The second fluid enters via opening 13, passes through center space of rotor and exits via openings 24 and exit 21. 16 is an end support wall. Heat transfer occurs through heat conductive bellows-like elements 17.

In FIG. 2 an end view of the unit of FIG. 1 is illustrated. 11 is housing, 20 and 21 are fluid openings, 23 is shaft, 18 is longitudinal vane, 17 is heat transfer element, and 10 is support base.

In FIG. 3 another form of the unit is illustrated, which is similar to the unit of FIG. 1 with some modifications. 30 is base supporting bearings 39 and 50, and shaft 40. First fluid enters via opening 38, passes between vanes 42 and then passes through passages between bellows-like heat exchange members 33 and baffles 32, and then passes between vanes 51 and exits via

opening 48. The second fluid enters via opening 49, passes between heat exchange members 33 and baffles 45, and then exits via openings 47 and exit 41. 34 are axial vanes near housing periphery, 46 are axial vanes near center, 43 and 44 are radial vanes in spaces between baffles 45 and heat exchange member 33, and 36 and 37 are vanes in spaces between heat exchange member 33 and baffles 33, and 35 is a typical layer of thermal insulation that may be applied primarily in the area shown in each bellow to enhance transport of each fluid in the indicated direction through the heat exchanger. If the fluid flow direction is changed, then the location of the thermal insulation also may be changed, or such insulation eliminated.

The operation of the heat exchanger is as indicated hereinbefore. Two fluid streams are passed through the heat exchanger, in heat exchange relationship with each other, and the heat exchanger is caused to rotate. Rotation of the heat exchanger is not mandatory, but such rotation improves the amount of heat transferred.

Referring to the unit of FIG. 1, the bellows like heat transfer elements are usually made such that the radial parts which are disc like, are closely together, with the spacing similar to that used for fins in finned tubing, for example. With such close spacing between the heat transfer elements, the first fluid, which is nearest to housing periphery, and which usually is the warmer fluid, fills the spaces between the heat transfer elements and by fluid friction is compelled to rotate with the heat exchanger. Since the colder second fluid is on the other side of the heat transfer element, first fluid is cooled, and due to density differentials, will be ejected from the space between said heat transfer elements 17, and being replaced by warmer fluid from the space between vanes 18, thus creating turbulence, and improving the amount of heat transfer. Similar conditions also prevail on the side occupied by the second fluid, where the second fluid is warmed by heat received from first fluid between the heat transfer elements 17, thus making the fluid between the heat transfer elements 17 warmer and lighter which then will be replaced by heavier colder fluid thus creating turbulence and improving heat transfer. Further, the heat conductive material of elements 17 which is usually metal, is normally thin, and thus the heat resistance of said elements is low, and all of the heat transfer area is what is commonly known as primary heat transfer surface.

The unit of FIG. 3, has baffles to force the two fluids to pass through a convoluted path near the heat transfer surfaces. Since all the fluid usually passes through these spaces between baffles and heat transfer elements, spaces between such baffles and elements must be made large enough for the fluid volume. These larger spaces may necessitate use of vanes as shown by 43 and 44 and also by 36 and 37, to control fluid movements and force the fluid to follow the rotating heat exchanger. Also, one may use thermal insulation in manner shown, to improve the flow of the fluids, by utilizing density changes in the fluids. Thus, it is possible, by use of insulation, and vanes radially, to reduce or eliminate losses within the heat exchanger passages. Also, depending of the temperature differentials available between the two fluids, there may be some pressure gain for the fluids within the heat exchanger, especially if high rotational speeds are used.

The two fluids are shown in the drawings to be in counterflow. Parallel flow may also be used, if desired. Also, various entry and exit arrangements for the two

fluids into and out of the heat exchanger, may be used. Further, in some arrangements, it may be desirable to have the heat exchanger elements 17 and 33 to be tapered, so that the elements change diameter in the axial direction. Such tapering has not been shown in the drawings, as it would be possible for anyone to do it, if desired. Also, the heat transfer elements may be formed from sheetmetal with radial corrugations, to provide radial obstructions equal to vanes. The baffles also may be similarly corrugated, forming radial passages similar to those formed by radial vanes.

For most applications, the rotational speed of the heat exchanger need not to be high. However, when desired, by constructing the housing and other parts sufficiently heavy, high speeds can be used. The operation and function of the heat exchanger components at high speeds is similar to that hereinbefore described.

Various other types of heat exchange member constructions may be used, if desired, in lieu of the bellows-like heat exchange members, such as finned tubing arranged in various ways, also radial discs stacked axially with spacers to form a finned member with fins on both sides similar to a finned tube with fins both inside and outside, and also heat pipes arranged to be in heat exchange relationship with both fluids. In all such arrangements, centrifugal force would be employed to assist in circulation of the two fluids and to improve heat transfer by creating turbulent conditions. Also, the vanes may be omitted in the form of FIG. 3, and vanes may be used with the form of FIG. 1. Further, the vanes 15 and 19 may be made in different radial lengths, to provide pumping action for the fluid thus eliminating the need for an external circulator, and similarly vanes 42 and 51 may be made in different radial lengths, as well as vanes 43 and 44, and vanes 36 and 37. By making the upstream vane longer, than the downstream vane, pressure is generated, thus pushing the fluid through the heat exchanger. Also, vanes may be curved in conventional manner if desired, to provide for fluid flow through the heat exchanger.

What I claim is:

1. A heat exchanger comprising:
 - a. a support for shaft;
 - b. a shaft rotatably mounted on said support;

c. a heat exchanger housing mounted on said shaft to rotate therewith, said housing having an entry and an exit for a first fluid and an entry and an exit for a second fluid, a first fluid and a second fluid within said housing, a heat exchanger element within said housing to exchange heat between said first fluid and said second fluid, with said heat exchanger element separating said first fluid and said second fluid, said heat exchanger element being a convoluted bellows type member with the folds of the bellows being transverse to the housing shaft said bellows type member having radially outwardly extending heat conductive members alternately connected at their outward ends to form a first set of folds and connected at their inward ends to form a second set of folds, said first and second set of folds forming a continuous fluid tight heat exchanger element separating the two fluids, said first set of folds being open inwardly toward the heat exchanger center and filled by one of said fluids, said second set of folds being open outwardly toward the housing and filled by one of said fluids, with the first and second fluids being on opposite sides of said heat exchanger element and within the folds of the convoluted bellows, with centrifugal force assisting in the circulation of the fluids within the folds due to density differentials provided by heat transfer through the heat conductive heat exchanger members.

2. The heat exchanger of claim 1 wherein the first fluid is on the outward side nearer to the periphery of said heat exchanger housing and wherein said first fluid is warmer than said second fluid.

3. The heat exchanger of claim 1 wherein said heat exchanger element is provided with a baffle transverse to the shaft of said heat exchanger and located between convolutions of the convoluted member to guide the fluid past the surfaces of the convoluted member.

4. The heat exchanger of claim 3 wherein the baffle is provided with a first means for assuring that said first fluid will rotate with said heat exchanger element.

5. The heat exchanger of claim 4 wherein the baffle is provided with a second means for assuring that said second fluid will rotate with said heat exchanger element.

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[54] DUAL ROTOR HEAT EXCHANGER

[76] Inventor: Michael Eskeli, 7994-41 Locke Lane,
Houston, Tex. 77042

[21] Appl. No.: 407,665

[22] Filed: Oct. 18, 1973

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 219,212, Jan. 20, 1972,
Pat. No. 3,791,167, and Ser. No. 386,207, Aug. 6, 1973,
abandoned.[51] Int. Cl.² F25B 9/00; F25D 9/00;
F28D 11/02[52] U.S. Cl. 62/402; 165/86;
415/64; 415/114; 415/178; 416/96 R[58] Field of Search 415/64, 178, 177, 179,
415/114, 199 A; 416/95, 96; 126/247; 62/401,
402, 403; 122/26; 165/86

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Primary Examiner—C. J. Husar

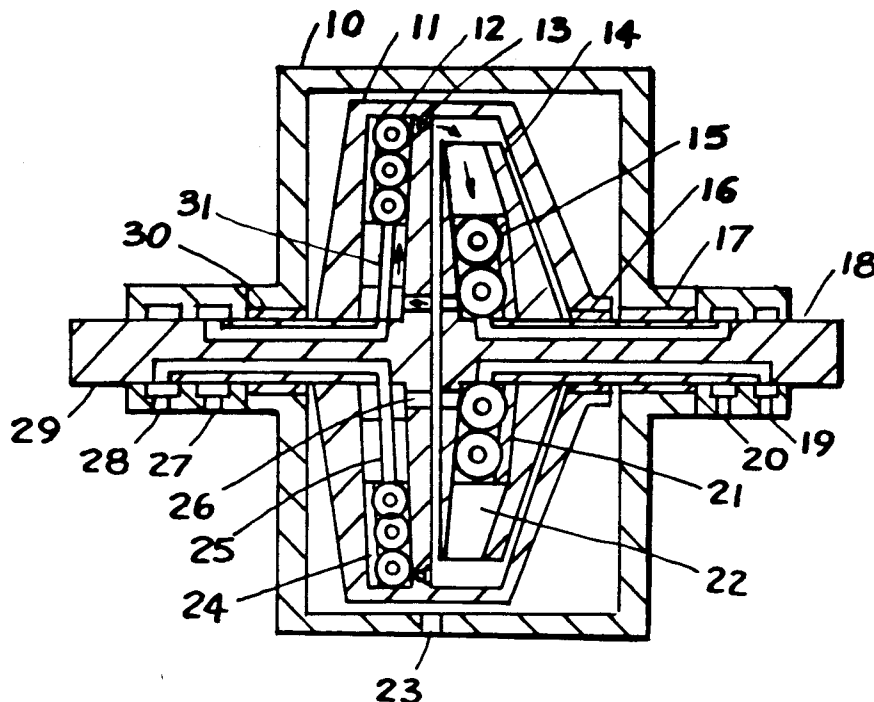
Assistant Examiner—Sheldon Richter

[57]

ABSTRACT

A method and apparatus for generating heat and vaporizing of liquids, and for providing cooling by using a continuous flow centrifuge to compress a gas with accompanying temperature increase and removing heat to another fluid from said gas in compressed state; said gas then being passed from the first rotor via nozzles mounted on said first rotor in backward direction thus reducing the absolute tangential velocity of said gaseous fluid; said gaseous fluid being then passed to a second rotor for which the rotor tip speed may be nearly the same as the velocity of said entering gaseous fluid. After said gaseous fluid enters the second rotor, it is then passed inward within said second rotor and discharged near said second rotor center. Work is supplied to said first rotor normally, and recovered in said second rotor. The gaseous fluid may be air, carbon dioxide, or some other fluid. The fluid receiving said heat may be water, ammonia, or some other fluid. A third fluid may be employed to add heat to said gaseous fluid during expansion; then said gaseous fluid is directly returned to said first rotor; said third fluid may be water, or be some other fluid.

3 Claims, 5 Drawing Figures



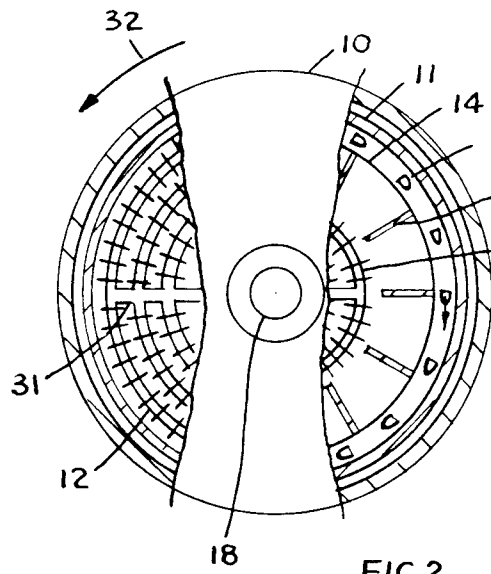


FIG. 2

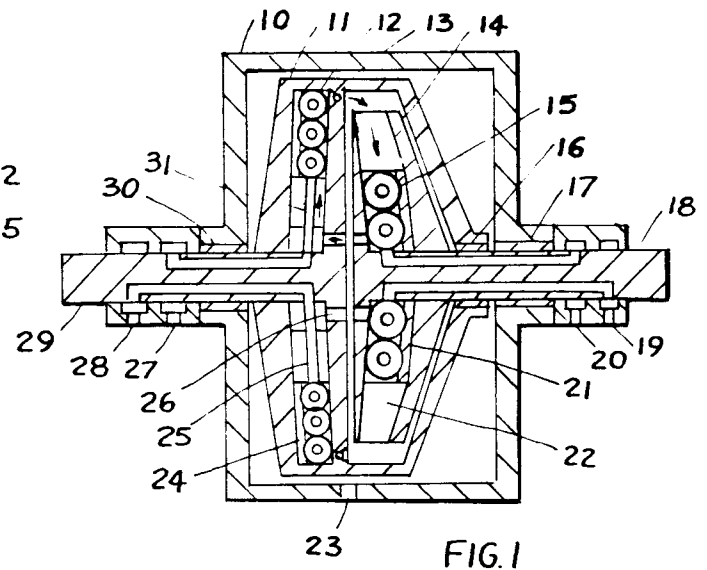


FIG. 1

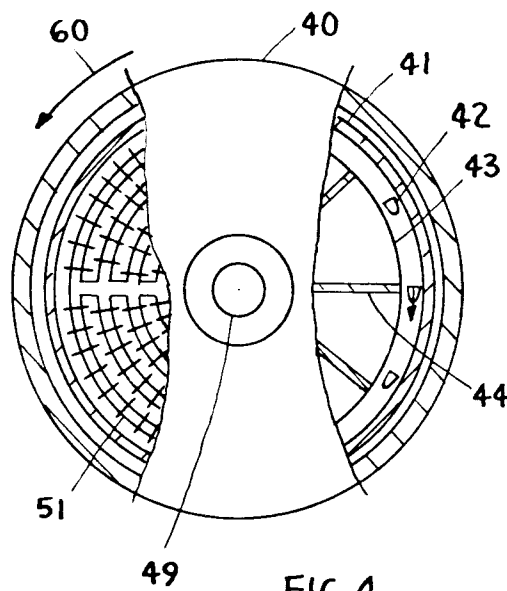


FIG. 4

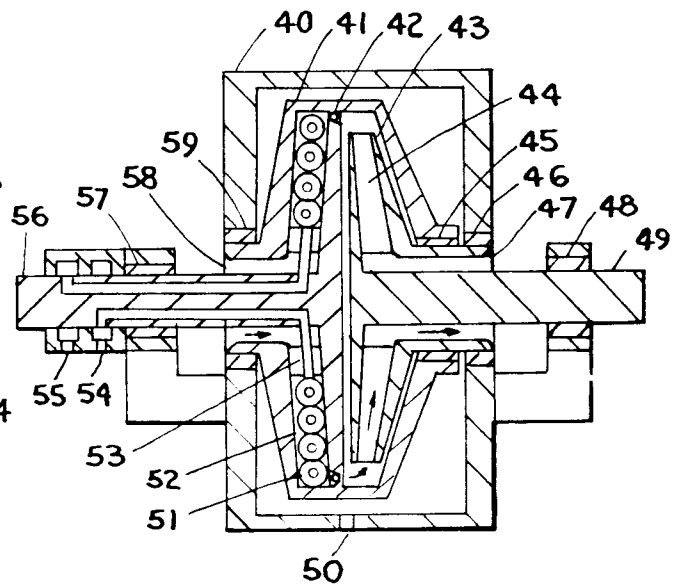


FIG. 3

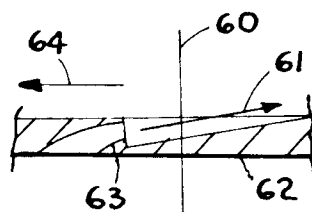


FIG. 5

DUAL ROTOR HEAT EXCHANGER

CROSS REFERENCES TO RELATED APPLICATIONS:

This application is a continuation-in-part of patent application "Rotary Heat Exchanger with Dual Rotors", filed Jan. 20, 1972, Ser. No. 219,212, now U.S. Pat. No. 3,791,167 and "Rotary Heat Exchanger with Dual Rotors", filed Aug. 6, 1973, Ser. No. 386,207 abandoned.

This invention relates generally to devices for generating heating and vaporizing fluids, and to devices for generating cooling, by compressing a gas within a continuous flow centrifuge and removing heat from said gas in a compressed state, and then adding heat to said gaseous first fluid after expansion of said first fluid.

Various types of heat pumps have been made in the past. These devices are generally inefficient and costly to construct.

FIG. 1 is a cross section of one form of the heat exchanger, and FIG. 2 is an end view of the unit shown in FIG. 1.

FIG. 3 is a cross section of another form of the device, and FIG. 4 is an end view of the unit of FIG. 3.

FIG. 5 is a detail of rotor nozzles.

It is an object of this invention to provide a method and apparatus for the generation of heating and cooling, and for vaporization of liquids, wherein a gas is compressed within a first rotor, and then discharged from nozzles mounted near the periphery of said first rotor for reduced work input to the heat exchanger, and for control of flow of said gaseous first fluid, with said first fluid then being passed to a second rotor for expansion and recovery of work, and with heat addition to said first fluid either within said second rotor or discharging said first fluid after said expansion.

Referring to FIG. 1, therein is illustrated a cross section of one form of the heat exchanger. 10 is casing, 11 is first rotor, 12 is first rotor heat exchanger, 13 are first rotor nozzles, 14 is second rotor, 15 is second rotor heat exchanger, 16 is seal, 17 is bearing, 18 is second rotor shaft, 19 and 20 are third fluid entry and exit to second rotor heat exchanger, 21 is second rotor heat exchanger support plate, 22 are second rotor vanes, 23 is opening to space within casing, 24 is first rotor heat exchanger support plate, 25 is first rotor vane, 26 is first fluid passage form second rotor cavity to first rotor cavity, 27 and 28 are second fluid entry and exit to first rotor heat exchanger, 29 is first rotor shaft, 30 is bearing, 31 is second fluid distribution conduit.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated with portions removed to show interior details. 10 is casing, 11 is first rotor, 14 is second rotor, 13 are first fluid nozzles, 22 are second rotor vanes, 15 is second rotor heat exchanger, 18 is second rotor shaft, 12 is first rotor heat exchanger, 31 second fluid distribution conduit, and 32 indicates direction of rotation for both rotors.

In FIG. 3, another form of the heat exchanger is shown in cross section. 40 is casing, 41 is first rotor, 42 is nozzle, 43 is second rotor, 44 are second rotor vanes, 45 is seal, 46 is seal, 47 is first fluid exit, 48 is bearing supporting second rotor shaft 49, 50 is opening to casing space, 51 is first rotor heat exchanger supported by plate 52, 53 is vane, 54 and 55 are entry and exit for second fluid to heat exchanger 51, 56 is first rotor shaft,

57 is bearing, 58 is first fluid entry to first rotor, 59 is seal.

In FIG. 4, an end view of the unit shown in FIG. 3 is illustrated with portions removed to show interior details. 40 is casing, 41 is first rotor, 42 are nozzles, 43 is second rotor, 44 are second rotor vanes, 49 is shaft, 51 is heat exchanger, and 60 indicates direction of rotation for both rotors.

In FIG. 5, a detail of rotor nozzles is shown. 60 is orientation of shaft about which the rotor rotates, 61 is direction of first fluid leaving nozzles 63, 62bis rotor dividing wall into which said nozzles are mounted, and 64 is direction of rotation of the rotor.

In operation, the first fluid is compressed within said first rotor by centrifugal action on said fluid by said rotor with accompanying temperature increase; this temperature increase provides the temperature differential to pass heat from said first fluid to said second fluid, with said second fluid being either a liquid or a gas with a temperature increase that is less than the temperature increase for said first fluid. After heat transfer, said first fluid is passed through said nozzles mounted near the periphery of said first rotor with said nozzles oriented to discharge said first fluid backward thus reducing the absolute tangential velocity of said first fluid. After leaving said first rotor nozzles, said first fluid enters said second rotor first fluid passages passing inward toward rotor center with vanes within said second rotor assuring that said first fluid will rotate with said second rotor, and said vanes recovering work from said first fluid when said fluid decelerates. In the unit shown in FIG. 3, the first fluid is passed from the second rotor via exit opening 47 near the center of said second rotor. In the unit shown in FIG. 1, heat is added to said first fluid during deceleration and expansion within said second rotor and then said first fluid is passed to the first rotor via openings near the center of both rotors. In the unit shown in FIG. 3, the first fluid is provided from external sources to said first rotor and enters via entry port 58 near the center of said first rotor.

The rotational tangential speed of the said second rotor is normally less, for many fluids, than the tangential speed for said first rotor, to provide for pressure differential required to circulate said first fluid through said rotors. This slower speed for said second rotor means work input to said heat exchanger unit, to circulate said first fluid. To reduce this work input, the nozzles 13 and 42 are provided. Work is supplied to said first rotor to accelerate said first fluid to the rotor tangential velocity, and some of this work is reclaimed in said first rotor nozzles when said first fluid leaves said nozzles in a backward direction. Additional work is then reclaimed in said second rotor. Normally, the shafts of the two rotors are connected by suitable power transmission device to pass the work obtained from said second rotor to said first rotor, with additional work being then supplied from external sources.

The first fluid velocity leaving said first rotor nozzles is usually low. Said nozzles are sized and shaped to obtain the highest attainable velocity for said first fluid relative to said nozzles for the pressure differential available, between entry and exit ends of said nozzles.

The casing of the heat exchanger may be evacuated to reduce fluid friction on external surfaces of the rotor.

Normally the first fluid radial velocity is low within each rotor, and said radial velocity may be controlled by providing suitably sized nozzles 13 or 42. The velocity of first fluid leaving said nozzles is dependent of the

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required velocity differential between first and second rotors; usually the tip velocity of said second rotor will be the same as the absolute tangential velocity of the first fluid entering said second rotor.

Various controls and governors may be used with the heat exchanger of this invention. They do not form a part of this invention and are not further described herein.

The starting point of the heat exchanger, as measured radially from center of rotation of said first rotor, may be as desired, and as required for the second fluid temperatures. The starting point has been shown different in FIGS. 1 and 3, for this reason; for a second fluid that is relatively cold at entry to heat exchanger 12 or 51, the heat exchanger will start near the rotor center for best efficiency with continuous heat exchange taking place til periphery. Generally, the second fluid and first fluid will both progress together from center to periphery both gaining in temperature, with said second fluid then being returned back from the periphery directly, to passage in rotor shaft.

The fluids used with the heat exchanger of this invention may be either liquids or gases. Said first fluid is normally a gas near the center of the rotors, but may be a liquid near rotor periphery. Alternately, said first fluid may be a suitable liquid at all points. Suitable fluids for use as said first fluid are air, nitrogen, carbon dioxide as gases, and many of the hydrocarbons, and fluids such as carbon disulfide, may be used as liquids. The temperature increase for said first fluid is always greater within said first rotor than for said second fluid. Said second

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fluid may be either a liquid or a gas. Fluids such as water, liquid ammonia, or other fluids may be used. For the third fluid, water or other fluids may be used.

What is claimed is:

1. A heat exchanger comprising a first rotor and a second rotor, means for mounting said rotors for independent rotation, a closed, first-fluid circulation path through said rotors, circulation path comprising a first passage extending approximately radially outwardly within said first rotor a second passage extending approximately radially outwardly at least partially within said second rotor, first fluid communication means communicating between the radially outward ends of said first and second passages, and second fluid communication means comprising first and second conduits through said first and second rotors respectively, communicating between the radially inward ends of said first and second passages, a heat removal exchanger carried by one of said rotors to remove heat from said first fluid being heated by centrifugal force, and a heat addition heat exchanger carried by one of said rotors to add heat to said fluid being cooled by expansion against centrifugal force.

2. The device of claim 1 wherein said heat removal heat exchanger is provided said first rotor, and said heat addition heat exchanger is provided said second rotor.

3. The device of claim 1 wherein said first communication means includes a nozzle mounted on said first rotor for discharging said first fluid into said second rotor.

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[54] **FLUID PRESSURIZER**
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Houston, Tex. 77042

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[21] Appl. No.: **754,734**

Primary Examiner—C. J. Husar

[22] Filed: **Dec. 27, 1976**

[57] **ABSTRACT**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 568,895, April 17,
1975, Pat. No. 4,003,673, and a continuation-in-part of
Ser. No. 636,310, Nov. 18, 1975, Pat. No. 4,012,164.

[51] Int. Cl.² **F01D 1/18**

[52] U.S. Cl. **415/80**

[58] Field of Search 415/80; 60/482;
416/179

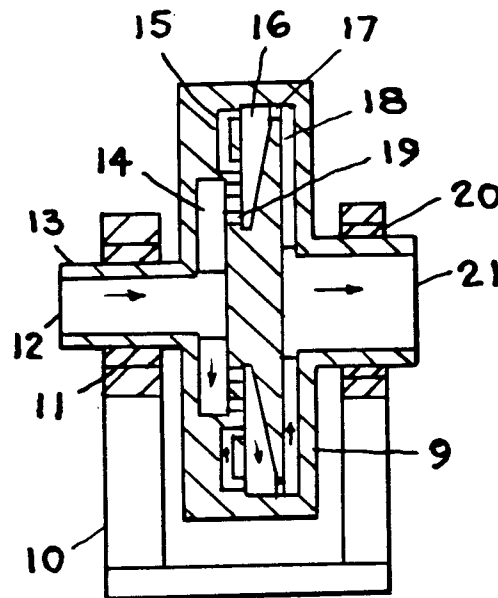
A method and apparatus for the pressurizing of fluids within a rotating rotor wherein a fluid enters the rotor near center, is first pressurized by centrifugal force with initial acceleration and is then further accelerated and pressurized within a circular cavity in a free vortex. The fluid then leaves the circular cavity via openings at periphery and is passed inwardly toward rotor center via passages, and discharged via opening near rotor center. Fluids may be gases or liquids. The device can be used either as a pump, a compressor, or a turbine as desired.

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2 Claims, 4 Drawing Figures



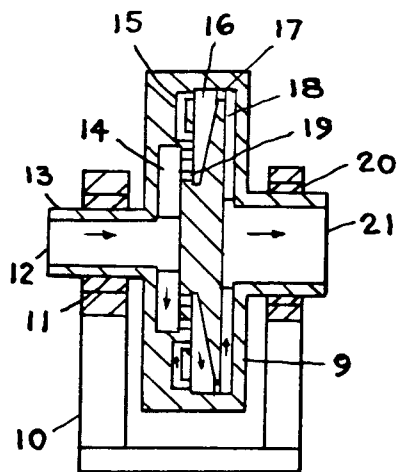


FIG. 1

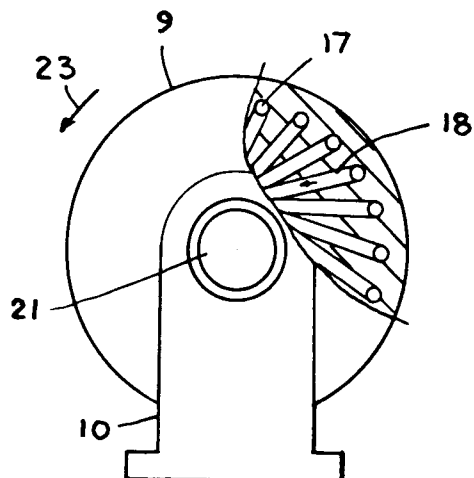


FIG. 2

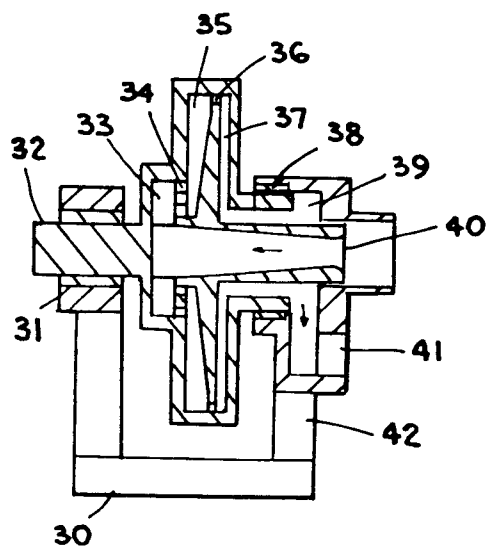


FIG. 3

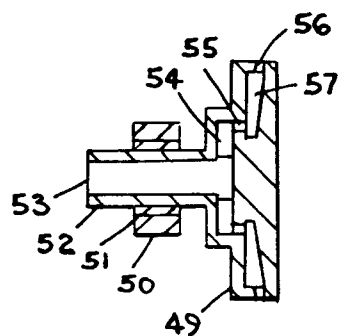


FIG. 4

FLUID PRESSURIZER

CROSS REFERENCES TO RELATED APPLICATIONS

This is a continuation-in-part of patent application "Fluid Pressurizer", Ser. No. 568,895, filed Apr. 17, 1975 now U.S. Pat. No. 4,003,673, and "Rotor with Recirculation", Ser. No. 636,310, filed Nov. 28, 1975 now U.S. Pat. No. 4,012,164.

BACKGROUND OF THE INVENTION

This invention relates to fluid pressurizers where a centrifugal force is used to increase the pressure of the fluid.

In previous fluid pressurizers of the centrifugal type, fluid is accelerated in the rotor, and then decelerated in the casing producing pressure. These methods are costly in power consumption due turbulence and friction losses, and for high pressures, the operation may become unstable, which is ordinarily corrected by adding stages, thus further increasing the cost of the unit.

SUMMARY OF THE INVENTION

It is an object of this invention to a means for pressurizing fluids with a reduced power requirement while still maintaining a relatively simple construction for the pressurizer. It is also an object of this invention to provide a rotor for such pressurizer which can be operated in multistage arrangements, or be used in situations where high rotor speeds are desirable.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial cross section of the pressurizer, and FIG. 2 is an end view of the unit of FIG. 1.

FIG. 3 is a cross section of a different arrangement of a unit similar to the unit shown in FIG. 1.

FIG. 4 is a cross section of a rotor for the pressurizer with fluid discharge at the rotor periphery.

Referring to FIG. 1, therein is shown an axial cross section of a pressurizer. 10 is base, 11 is bearing, 12 is fluid entry, 13 is shaft, 14 is pressurizing cavity provided with vanes, 15 is recirculation passage, 16 is free vortex cavity, 17 are fluid exit openings which may be nozzles, 18 are fluid inward passages, 19 are fluid entry nozzles into free vortex cavity, 20 is bearing, 21 is fluid exit opening, 9 is rotor.

In FIG. 2, an end view of the unit of FIG. 1 is shown. 10 is base, 21 is fluid exit, 23 indicates rotation, 9 is rotor, 17 are fluid exits from vortex cavity, 18 are inward passages.

In FIG. 3, a cross section of another form of the unit is shown, where entry and exit for the fluid are on the same side of the rotor. 30 is base, 31 is bearing, 32 is shaft, 33 is pressurizing cavity, 34 are nozzles, 35 is free vortex cavity, 36 are openings for fluid exit, 37 are inward passages, 38 is seal, 39 is fluid exit passage into exit 41, 40 is fluid entry which may be tapered as shown, 42 is support for the fluid connections assembly.

In FIG. 4, a rotor is shown; this rotor may be used within a vessel, in the open, or in multistage arrangements. 49 is rotor, 50 and 51 are support and bearing, 52 is shaft, 53 is fluid entry, 54 is pressurizing cavity, 55 are fluid nozzles into free vortex cavity 57, and 56 are exit openings or nozzles for the fluid.

In operation, fluid enters via opening 12, is pressurized by centrifugal force in cavity 14 where vanes are used to assure that the fluid rotates with the rotating

rotor, after which the fluid passes via nozzles 19 forwardly into free vortex cavity 16, and there is pressurized by centrifugal force while traveling within the curved path formed by cavity 16. After pressurization, the fluid leaves via openings or nozzles 17 and is passed inwardly via passages 18 to exit 21. The passages 18 may be radial, or they may be arranged to be either forwardly or backwardly as desired; in FIG. 2, the passages 18 are forwardly to allow the fluid to lead the rotor, and thus retain additional pressure while passing toward rotor center. Work is recovered from the fluid in passages 18, the amount of work recovered will depend of the design of the slope of the passages 18. The passage 15 is used to recirculate the fluid taking fluid from the free vortex cavity at periphery, and discharging via nozzles near center; purpose of this recirculation is to increase the fluid tangential velocity within the rotor free vortex cavity.

The openings 56 may be nozzles arranged to discharge the fluid backwardly, if desired. For the rotor of FIG. 4, various arrangements for supporting shaft may be used as desired.

Generally, the apparatus disclosed herein is intended for high speed operation, and by eliminating the surrounding stationary casing, friction losses are reduced for the rotor. Such friction losses are high for fluids such as water, where water is at a high pressure. Friction losses within the rotor of this device are low, due to the normally low velocity differentials within the rotor between the fluid and the rotor.

Applications for this pressurizer include pumping of liquids, and as a gas compressor.

The function of the recirculation passages 15 were disclosed in a previous patent application "Rotor with Recirculation", Ser. No. 636,310, filed Nov. 28, 1975.

The free vortex cavity is shaped as a circular path, generally transverse to the rotor axis, and the working fluid injected into the cavity tangentially, with speed that is the sum of the local rotor speed at the entry nozzles, and the fluid tangential velocity leaving the nozzles. The fluid decelerates relative to the rotor as it travels outwardly, and normally the rotor tangential speed and the fluid tangential speed are the same at rotor periphery. The fluid pressure is usually nearly nil at the center of the free vortex cavity, and increases with increasing radius due to the centrifugal force of the rotating fluid. The free vortex cavity is usually tapered as shown to allow for the maintenance of the required high velocities within the cavity relative to rotor. Normally, the fluid entry pressure to the unit is elevated, and additional pressure is generated in the pressurizing cavity, so that the leaving velocity of the fluid relative to nozzles when entering the free vortex cavity is the result of the pressure drop entry plus centrifugal pressure in pressurizing cavity, to nil. By this means, the needed high fluid velocities in the free vortex cavity are obtained.

I claim:

1. A fluid pressurizer comprising:

- a. a shaft mounted for rotation;
- b. a rotor mounted on said shaft to rotate therewith, said rotor being provided with an entry for a fluid to be pressurized, said entry communicating with a pressurizing cavity extending outwardly from the rotor center, said pressurizing cavity communicating with a set of feeder nozzles for passing a fluid forwardly in the direction of rotation into a free vortex cavity of circular configuration and trans-

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verse to the rotor axis, said free vortex cavity communicating near its periphery with exit openings to allow the discharge of the fluid from the free vortex cavity.

is further provided with a set of inwardly extending passages for the fluid for passing the fluid into an exit near the rotor center.

2. The fluid pressurizer of claim 1 wherein said rotor

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United States Patent [19]

Eskeli

[11] 4,077,230

[45] * Mar. 7, 1978

[54] **ROTARY HEAT EXCHANGER WITH COOLING**

[76] Inventor: **Michael Eskeli**, 7994-41 Lacke Lee, Houston, Tex.

[*] Notice: The portion of the term of this patent subsequent to Jan. 20, 1993, has been disclaimed.

[21] Appl. No.: **474,729**

[22] Filed: **May 30, 1974**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 361,281, May 17, 1973, abandoned.

[51] Int. Cl.² **F25B 3/00**

[52] U.S. Cl. **62/401; 62/499; 62/501; 415/1**

[58] Field of Search **62/401, 499, 500, 501; 415/1**

[56] References Cited

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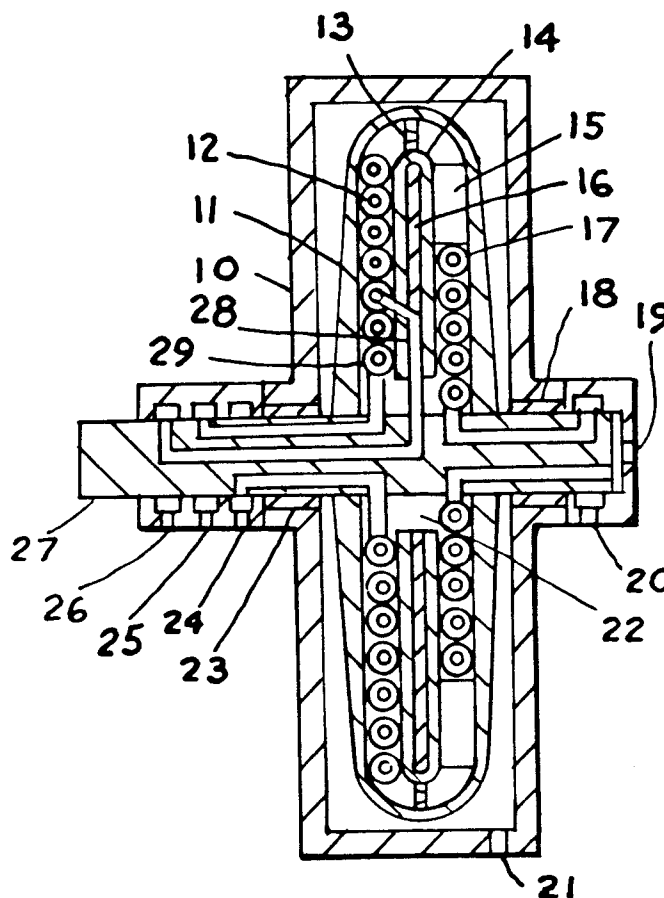
Primary Examiner—Charles J. Myhre

Assistant Examiner—Sheldon Richter

[57] ABSTRACT

A method and apparatus for the transfer of heat at a lower temperature to another fluid at a higher temperature, using a rotary heat exchanger and circulating the two fluids through said heat exchanger wherein a third fluid is circulated. The third fluid is normally a gas, compressed within the rotor by centrifugal action, with accompanying temperature increase, and heat is removed from said third fluid to a second fluid during and after compression; heat is added to said third fluid from a first fluid during and after expansion. A fourth fluid may be also circulated within said rotor, for removing heat from said third fluid before and during early part of compression to increase the weight of said third fluid within the compression side of the said rotor, thus improving the circulation of said third fluid within said rotor. Said second fluid, said first fluid, and said fourth fluid may be either liquids or gases as desired, including water. Said third fluid may be carbon dioxide.

3 Claims, 5 Drawing Figures



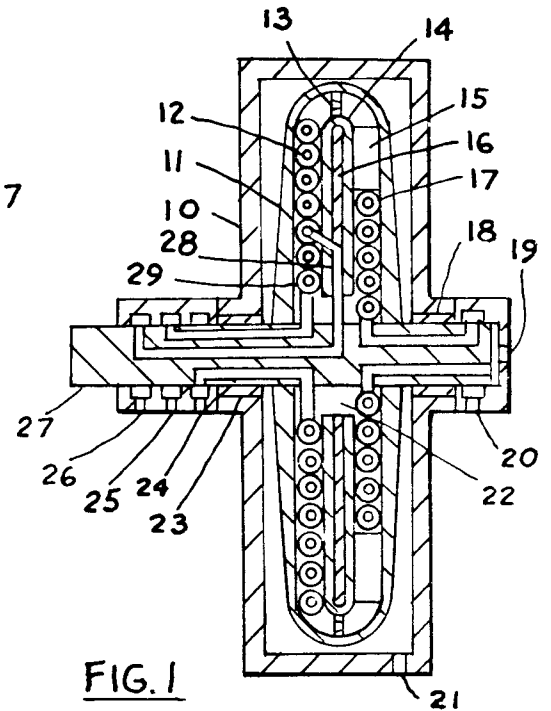
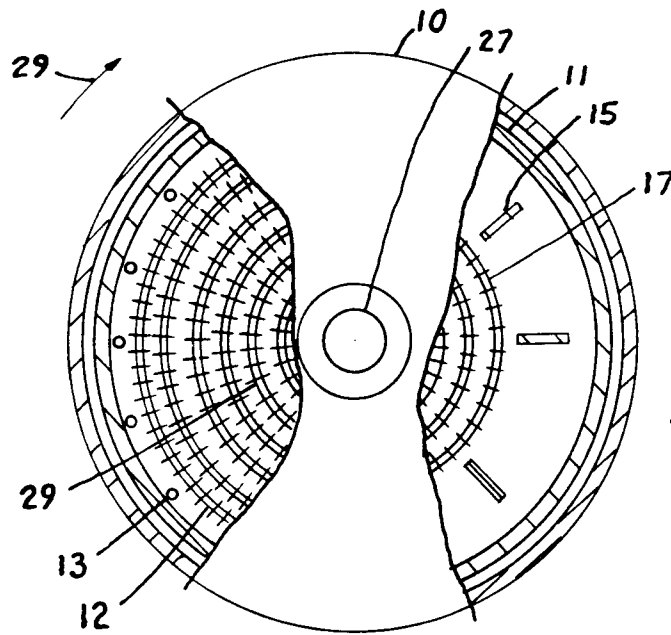


FIG. 2

FIG. 1

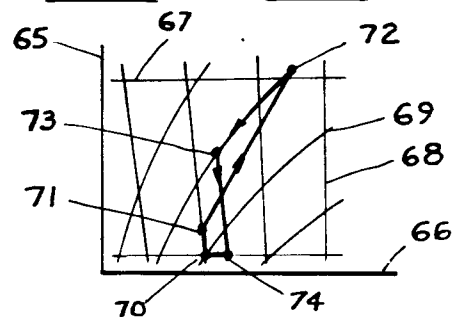


FIG. 5

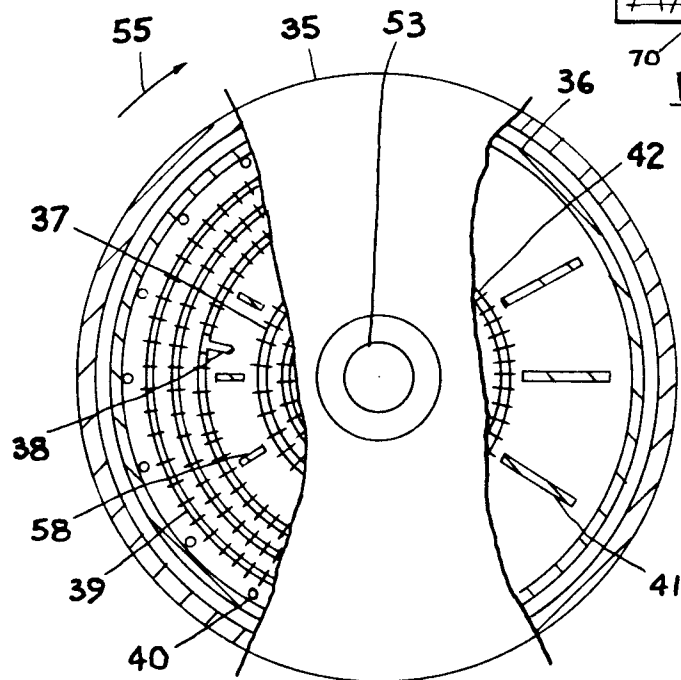


FIG. 4

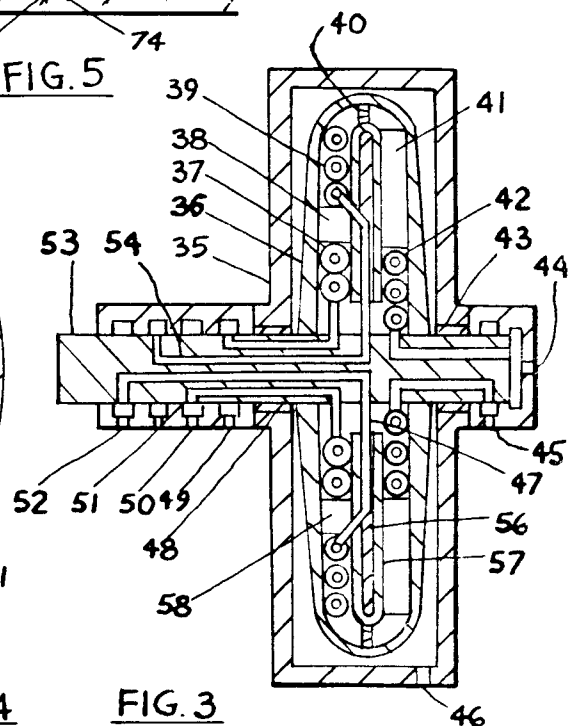


FIG. 3

ROTARY HEAT EXCHANGER WITH COOLING

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a Continuation-in-part application, of "Heat-Exchanger with Three Fluids", filed May 17, 1973, Serial No. 361,281, now abandoned.

BACKGROUND OF THE INVENTION

This invention relates to devices for producing heating and cooling, wherein one fluid releases heat and is thus cooled, and another fluid gains heat and is thus heated.

DESCRIPTION OF THE PRIOR ART

The art of producing heating and cooling has seen a variety of devices. In some of those devices, heat is generated by burning a fuel within a device, and the heat produced by this chemical reaction is transferred to a fluid being circulated in conduits in heat exchange relationship with said heating substance.

The main disadvantage of these conventional systems is that they require large amounts of fuel to generate the needed high temperatures to vaporize a fluid, and also require a large amount of power to produce a predetermined amount of cooling.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 is a cross section of a form of the device for producing said heating and

FIG. 2 is an end view of the unit shown in FIG. 1.

FIG. 3 is a cross section of another form of the device, and FIG. 4 is an end view of the unit shown in FIG. 3.

FIG. 5 is a pressure enthalpy diagram for a typical third fluid with the work cycle for said fluid superimposed thereon.

DESCRIPTION OF PREFERRED EMBODIMENTS

It is an object of this invention to provide means for removing heat from a third fluid being circulated within a rotor before and during compression thus providing for increased density for said third fluid and thus increasing the weight of said third fluid within the compression side of said rotor, which may be necessary for many fluids when used as said third fluids for increased capability of heat removal during and after compression from said third fluid into a second fluid.

Referring to FIG. 1, therein is shown a cross section of a form of the device, where 10 is casing, 11 is rotor, 12 is second fluid heat exchanger, 13 are openings or nozzles, 14 is rotor divider, 15 is vane on rotor expansion and deceleration side, 16 is thermal insulation within rotor divider, 17 is first fluid heat exchanger, 18 is rotor shaft bearing and seal, 19 is first fluid exit, 20 is first fluid entry, 21 is casing vent, 22 is third fluid passage near rotor center, 23 is rotor shaft bearing and seal, 25 is second fluid entry, and 24 and 26 are second fluid exits, 27 is rotor shaft, 29 indicates a portion of second fluid heat exchanger used for cooling said third fluid, and 28 is second fluid conduit.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated, with portions removed to show internal details. 10 is casing, 11 is rotor, 27 is rotor shaft, 15 is vane, 17 is first fluid heat exchanger, 12 is second fluid heat exchanger, 13 are third fluid passage openings or

nozzles, 29 is portion of second fluid heat exchanger, 28 indicates conduit for discharging a portion of said second fluid, 29 indicates direction of rotation of rotor.

In FIG. 3, a cross section of another form of the device is shown. 35 is casing, 36 is rotor, 37 is cooling heat exchanger for removing heat from third fluid before and during early part of compression by circulating a fourth fluid within said heat exchanger 37, 38 is second fluid conduit, 39 is second fluid heat exchanger, 40 is third fluid passage opening or nozzle, 41 is vane, 42 is first fluid heat exchanger, 43 and 48 are shaft bearings and seals, 44 and 45 are first fluid entry and exit, 56 is thermal insulation within rotor divider wall 57, 46 is casing vent, 58 is vane, 47 is third fluid passage from deceleration side to acceleration side, 49 and 50 are fourth fluid entry and exit, 51 and 52 are second fluid entry and exit, and 54 is second fluid conduit within rotor shaft.

In FIG. 4, an end view of the unit shown in FIG. 3, is illustrated. 35 is casing, 53 is rotor shaft, 36 is rotor, 42 is first fluid heat exchanger, 41 is vane, 40 is third fluid opening or nozzle, 39 is second fluid heat exchanger, 58 are vanes, 38 is second fluid conduit, 37 fourth fluid heat exchanger, 55 indicates direction of rotation of rotor.

In FIG. 5, a pressure-enthalpy diagram for a typical third fluid is illustrated, with 65 being the pressure line and 66 being the enthalpy line, 67 are constant pressure lines, 68 are constant temperature lines, and 69 are constant entropy lines. The working cycle of said third fluid is superimposed thereon, and compression is shown from 70 to 71 approximately isothermally with said fourth fluid providing the necessary cooling, from 71 to 72 the compression is with some cooling, and said second fluid provides here the cooling, expansion is from 72 to 73 isentropically, and from 73 to 74 with heat addition with heat being provided from said first fluid, and then from 74 to 70 with cooling, where said cooling is provided by said fourth fluid.

In operation, said third fluid is compressed by centrifugal force within said rotating rotor, with accompanying temperature increase. During said compression, during later part of said compression, heat is removed from said third fluid to said second fluid with said second fluid being circulated within said second fluid heat exchanger in heat exchange relationship with said third fluid. After compression and said heat transfer, said third fluid is decelerated with accompanying pressure and temperature decrease with suitable vanes assuring that said third fluid will rotate with said rotor for recovery of work associated with said deceleration. During latter part of said deceleration, after suitable reduction of said temperature of said third fluid, heat is added to said third fluid from said first fluid being circulated in heat exchange relationship with said third fluid. After said heat addition, said third fluid is passed again to be accelerated and thus compressed. During early part of said compression, heat is removed from said third fluid by circulating a fourth fluid within a fourth fluid heat exchanger in heat exchange relationship with said third fluid; also, heat may be removed by said fourth fluid from said third fluid before said compression. After further acceleration and compression, heat is transferred to said second fluid, thus completing the work cycle for said third fluid.

An alternate method of providing cooling for said third fluid before and during early part of said compression is shown in FIG. 1, where a part of said second fluid is employed as a coolant for said third fluid. A

portion of said second fluid is discharged via conduit 28, and the remainder of said second fluid then continues to receive heat and is then discharged as the heated second fluid via another exit 24.

Cooling of the said third fluid from 74 to 71, FIG. 5, is often required to balance the third fluid weight on the acceleration and deceleration sides of said rotor, so that the weight of said third fluid is greater on the acceleration and compression side of said rotor than on said deceleration and expansion side of said rotor. By providing additional cooling before and during early part of said compression, the third fluid density is increased and thus its weight is increased during said compression. This is necessary with many gaseous fluids when used as said third fluid, and in particular, it is necessary when the amount of said second fluid being circulated is relatively small during the latter part of said compression of said third fluid. Normally, the amount of said second fluid circulated is small when high final temperatures of said second fluid are desired, and when the amount of said third fluid being circulated is large when compared to the amount of said fluid being circulated.

Further, when the form shown in FIG. 3 is used, the entering temperature of said second fluid may be high, since the necessary cooling for said third fluid is provided by said fourth fluid.

The rotor is normally made of high strength materials to provide for high speed capability. The heat exchangers may be made of finned tubing as shown. The casing is usually evacuated to provide for elimination of drag on rotor outer surfaces. Power is provided to rotor shaft to rotate said rotor from an external source.

The third fluid is sealed within said rotor and is circulated therein by providing for placement of said heat exchangers such that the weight of said third fluid is greater on the said acceleration side of said rotor than on said deceleration side of said rotor, thus creating a pressure differential, necessary for transporting said third fluid within said rotor.

The said third fluid is normally a gas, such as carbon dioxide, or some other gas. The said second fluid, first fluid and fourth fluid may be either a gas or a liquid; typical fluid is water. Heat is supplied by said first fluid into said third fluid, and then said heat is transferred at a higher temperature to said second fluid, and from there to a use point. Work is supplied to accelerate said fluids within said rotor, and work is recovered when said fluids decelerate within said rotor.

The said first fluid, second fluid and said fourth fluid, are supplied from external sources, at suitable temperatures.

In FIG. 1, item 13 is a nozzle or opening in the divider for passing said third fluid from the compression side of said rotor to expansion side of said rotor. These passages 13 may be plain openings, or they may be used to regulate the flow of said third fluid within said rotor. Further, said passages may be made into nozzles oriented to discharge said third fluid in a desired direction, which may be backward or forward. When discharging said third fluid backward away from direction of rotation, the absolute tangential velocity of said third fluid is reduced, and thus the pressure on the expansion side of the rotor is reduced thus assisting in the circulation of said third fluid. When said passages 13 are nozzles oriented to discharge said third fluid forward to the direc-

tion of rotation, the absolute tangential velocity of said third fluid is increased, and this velocity increase will result in an increased work output by the expansion side of said rotor, thus reducing the work input to rotor shaft from external sources. The use of the passages 13 as nozzles, is dependent of the temperature differentials available between the coolant fluid which is said fourth fluid or entering second fluid, and the heating fluid which is said first fluid. For a close temperature differential between said fluids, said nozzles may be arranged for backward discharge, and for a large temperature differential, said nozzles may be arranged to discharge forward thus reducing or eliminating the need for an external power source to rotate said rotor.

Various controls and governors may be used with the device of this invention. They do not form a part of this invention and are not further described herein.

I claim:

1. In a rotary heat exchanger, comprising a rotatably mounted rotor, said rotor having first and second passages extending continuously outwardly from the axis of rotation of said rotor, passage means for connecting the outer ends of said first and said second passages, passage means for connecting the inner ends of said first and said second passages, to allow a fluid to flow outwardly and be compressed in said first passage and to flow inwardly and expand toward the center of rotation in said second passage, a compressible fluid within said passages, a first heat exchanger carried by said rotor for removing heat from said fluid and being at least in part adjacent to the outer end of said first passage and being at least in part adjacent to the outer end of said second passage, a second heat exchanger carried by said rotor and located in the second passage for adding heat into said fluid, the improvement comprising a third heat exchanger carried by said rotor adjacent to the inner ends of said first passage, for removing heat from said fluid as it enters the first passage.

2. The rotary heat exchanger of claim 1 wherein said third heat exchanger is a part of said first heat exchanger being in the part of said first heat exchanger nearest to the inner ends of said first and said second passages.

3. In a rotary heat exchanger, comprising a rotatably mounted rotor, said rotor having first and second passages extending continuously outwardly from the axis of rotation of said rotor, passage means for connecting the outer ends of said first and said second passages, passage means for connecting the inner ends of said first and said second passages, to allow a fluid to flow outwardly and be compressed in said first passage and to flow inwardly and expand toward the center of rotation in said second passage, a compressible fluid within said passages, a first heat exchanger carried by said rotor for removing heat from said fluid and being at least in part adjacent to the outer end of said first passage and being at least in part adjacent to the outer end of said second passage, a second heat exchanger carried by said rotor and located in the second passage for adding heat into said fluid, the improvement comprising a third heat exchanger carried by said rotor adjacent to the inner ends of said first passage for removing heat from said fluid downstream from said second heat exchanger.

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United States Patent [19]

[11]

4,106,304

Eskeli

[45]

Aug. 15, 1978

[54] THERMODYNAMIC COMPRESSOR

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee,
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[21] Appl. No.: 724,407

[22] Filed: Sep. 17, 1976

Primary Examiner—Charles J. Myhre

Assistant Examiner—Sheldon Richter

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 708,863, Jul. 26, 1976,
Ser. No. 546,912, Feb. 4, 1976, abandoned, and Ser.
No. 600,312, Jul. 30, 1975, Pat. No. 3,986,361.

[51] Int. Cl.² F25B 3/00

[52] U.S. Cl. 62/88; 60/650;
62/402; 123/220; 165/DIG. 12; 418/86

[58] Field of Search 62/401, 86, 88, 402;
60/650, 682; 418/86; 417/243; 415/178;
165/DIG. 12

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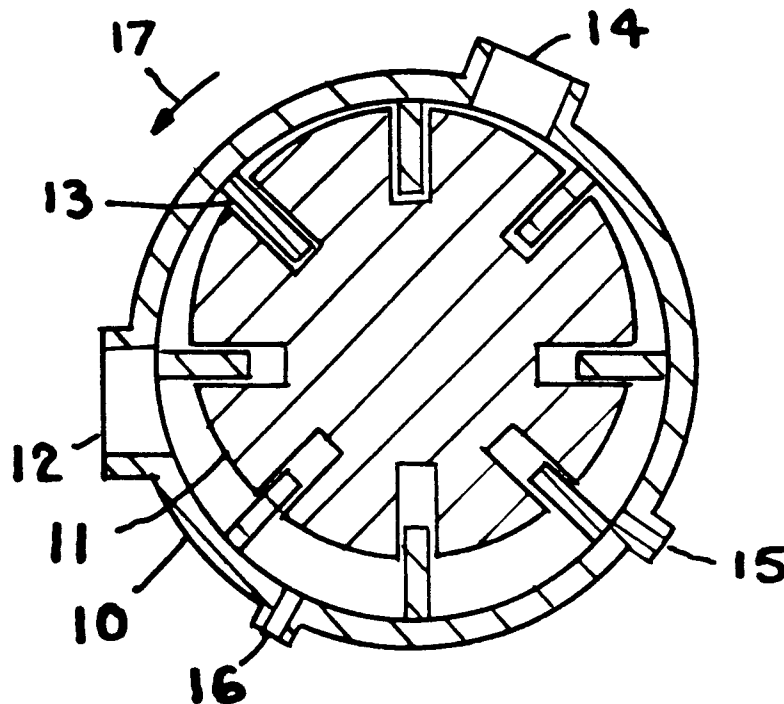
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[57] ABSTRACT

A method and apparatus for the compression of compressible fluids wherein a fluid is first compressed within a constant volume with addition of heat, and then further compressed in a compressor means increasing the fluid pressure and temperature. After compression, the fluid is passed into a heat exchanger for heat removal, with the heat so removed usually being used as the heat being added into the fluid in the initial step. After passing through the heat exchanger, the fluid leaves the process. Various types equipment may be used, including vane, piston, screw or other positive displacement type apparatus. In an alternate arrangement, heat may also be removed during the compression in the compressor means. Also, work may be extracted from the fluid after leaving the heat exchanger heat removal.

4 Claims, 5 Drawing Figures



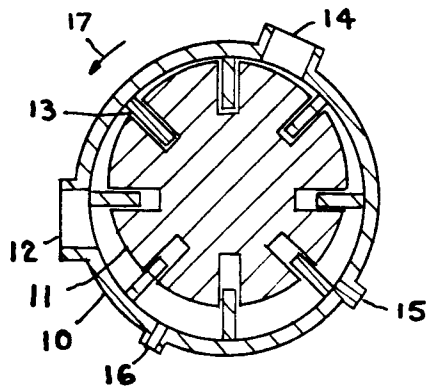


FIG. 1

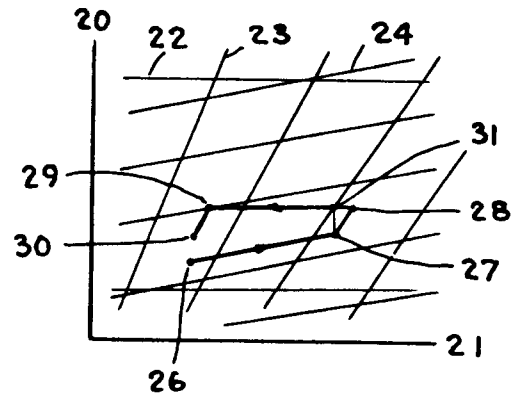


FIG. 2

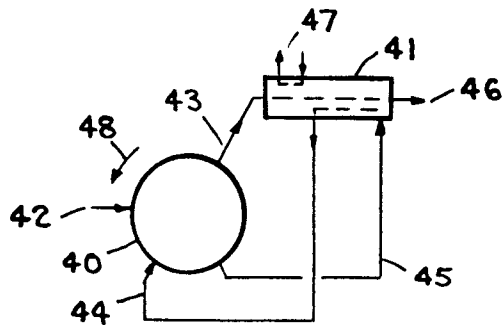


FIG. 3

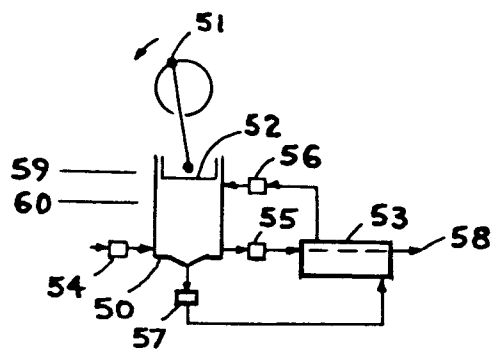


FIG. 4

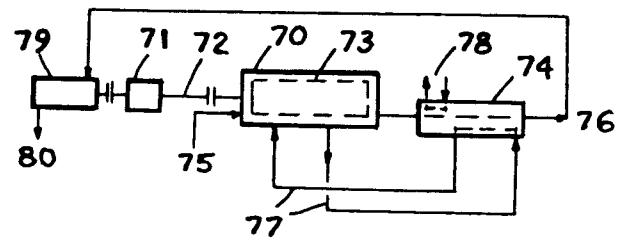


FIG. 5

THERMODYNAMIC COMPRESSOR

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part application of "Thermodynamic Compressor", Ser. No. 708,863, filed July 26, 1976, and "Power Generator", Ser. No. 546,912, filed February 4, 1976, now abandoned and "Turbine with Regeneration", Ser. No. 600,312, filed July 30, 1975, now U.S. Pat. No. 3,986,361.

BACKGROUND OF THE INVENTION

This invention relates generally to apparatus for compressing gases and other compressible fluids, wherein the fluid is heated first essentially at constant volume, and then work is added to the fluid compressing it, and after that heat is removed, with such removed heat being usually used to heat the fluid.

SUMMARY OF THE INVENTION

The object of this invention is to provide a means for compressing gases with reduced work input, by doing part of the gas compression at constant volume, with heat addition, thus increasing the gas pressure without shaft work. The heat required to be added is obtained regeneratively from the same fluid. Alternately, external heat may be used. Also, some heat at high temperature may be removed, and passed to other uses.

BRIEF DESCRIPTION OF THE DRAWINGS

In FIG. 1, a vane type unit is shown in detail, illustrating working fluid entry and exit opening approximate locations, and also heat transfer fluid entry and exit openings.

In FIG. 2, a typical pressure-internal energy diagram for the compression process is shown.

In FIG. 3, a typical system using the unit of FIG. 1, is shown.

In FIG. 4, a typical system using a piston and cylinder arrangement is shown, employing a work cycle as shown in FIG. 2.

In FIG. 5, a typical system using a screw type compressor is shown.

DESCRIPTION OF PREFERRED EMBODIMENTS

The systems shown in the several Figures use all positive displacement equipment for compressing means, and use a stationary heat exchanger to remove heat from the fluid after compression; this heat exchanger is similar to the well known after cooler used with commercial air compressors. However, instead of wasting the heat removed from the working fluid after compression, the heat is used to partially compress the working fluid by adding such heat into the working fluid at constant volume before actual mechanical compression. Thus, thermodynamic methods are used to pressurize the working fluid in part, avoiding the addition of work from external sources for that part of the compression.

In normal practice in the past, it has been a cardinal rule that for minimum work input one should have a working fluid as cold as possible at start of compression. However, one can reduce work input for the compression also by heating, if such heating is done in a confined space at constant volume, so that the fluid pressure in the volume is increased without work. In the arrange-

ment used here, work is added to the working fluid after such heat addition; such work addition also increases the working fluid temperature even further, and it is this work addition which maintains the process, and maintains the working fluid temperature at a predetermined level. Theoretically, one thus can set the temperature at start of mechanical compression as high as desired, and for best results, a high temperature, to the upper limits of what the materials of the compressor can stand, are desirable. At the same time, the actual work input to the working fluid by the mechanical compressor, can be almost as low as desired; typically, the work input to the working fluid should be perhaps 5 to 10 BTU/lb, and such work input is only in such instant intended to maintain the working fluid temperature, and not intended to compress the working fluid except in a minor way.

The process can be further explained with the help of the several Figures. In FIG. 1, a vane unit is shown, which is a normal vane unit with ports provided in suitable places for the entry and exit of the working fluid, and the entry and exit of a heat transfer fluid needed for heat addition to working fluid. 10 is casing, supporting and housing rotor 11, and vanes 13. 12 is working fluid entry, and 14 is working fluid exit. 15 is heat transfer fluid exit and 16 is heat transfer fluid entry, 17 is arrow.

In FIG. 2, a pressure-internal energy or enthalpy diagram is shown for a typical working fluid, with a work cycle for the compression process illustrated thereon. 20 is pressure line, 21 is internal energy or enthalpy line, 22 is constant pressure line, 23 is constant entropy line, 24 is constant volume line, line 26-27 is constant volume heat addition, 27-28 is isentropic compression, 28-29 is constant pressure heat removal, and 29-30 is an optical work removal from the working fluid, with point 30 located as desired at the desired pressure, and line 27-31 is a constant temperature compression process, which may be used alternately with the isentropic compression. Line 28-31 may be also used to indicate high temperature heat removal, if such is used, with the heat being passed to a use external to this process.

In FIG. 3, a system using the vane unit shown in FIG. 1 is shown, to work in accordance to the work cycle shown in FIG. 2. The rotation is indicated by 48, 42 is working fluid entry, 40 is vane unit, 44 and 45 are heat transfer fluid conduits, 46 is working fluid exit, 41 is heat exchanger, 47 is high temperature removal heat transfer fluid conduits, 43 is working fluid exit from vane unit.

In FIG. 4, a system is shown using a piston and cylinder arrangement. 50 is cylinder, 51 is crank, 52 is piston. working fluid enters via valve 54 and leaves via valve 55 and passes into heat exchanger 53 and exits via 58. Heat transfer fluid enters via valve 56 when piston is in position shown, which is also indicated by numeral 59. When piston is in position shown by line 60, heat transfer fluid leaves via valve 57, and passes through heat exchanger 53 to valve 56. The remaining compression of working fluid from line 60 to bottom end of piston travel, is the compression indicated by line 27-28 in FIG. 2; such piston travel also includes working fluid delivery out from the cylinder. The cylinder is then filled via valve 54 during the piston travel from cylinder bottom to cylinder top.

In FIG. 5, a unit using a screw type compressor is shown. 70 is screw unit casing and 73 is screw, con-

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 nected to drive 71 via shaft 72. Working fluid enters via 75, and during its passage through screw unit is heated by heating fluid passing through conduits 77, the working fluid then passes through heat exchanger 74 where heat is first removed to fluid in conduits 78 and then to heat transfer fluid in conduits 77, and after this, the working fluid may be discharged, or be passed into an expander 79, and then discharged via 80. The work extraction in the expander 79 corresponds to the work removal shown by line 29-30 in FIG. 2.

In operation, and referring to the system of FIG. 3, the compressor rotor is caused to rotate. The working fluid enters and fills the rotor cavity and then the vanes seal the packet of working fluid from the entry opening. The working fluid is then heated, in the approximately constant volume, by injecting a heat transfer fluid into the space, usually through suitable spray nozzles, so as to assure good mixing of the working fluid and the heat transfer fluid. The heat transfer fluid, in many instances may be light oil, providing also lubrications for the vanes. The heat transfer fluid is then removed before starting of mechanical compression, and the working fluid is compressed, with accompanying temperature increase, and then discharged to a stationary heat exchanger where the heat is removed from the working fluid and transferred into the heat transfer fluid to be used to heat the working fluid before mechanical compression. Heat can be also removed during mechanical compression by injecting a second heat transfer fluid into the rotor cavity and then removing it after warming of the second heat transfer fluid; this is not shown in drawings for the apparatus, but is indicated in FIG. 2, line 27-31.

The operation of the systems shown in FIGS. 4 and 5 is similar to the operation of the system of FIG. 3.

The applications for this system and method are in the compression of various gases with low energy consumption. The systems can be used to provide compression stages for various other apparatus, such as gas turbines, heat pumps and the like. Further, the system shown in the several figures may be made to include several stages, so that after the working fluid leaves at point 29, of FIG. 2, it again enters another system at its point 26, and undergoes a similar process gaining again in pressure.

The heat addition at constant volume can be accomplished using other means, in place of the heat transfer fluid being used herein, to add the heat. One can use jackets around the cylinders, for example, as is done in commercially available units. Similarly, jackets can be provided for the vane or screw units. One can also use heat exchangers placed within the mechanical compressor to be in heat exchange relationship with the confined working fluid, to provide heat addition at constant volume. All these devices are known to people in the trade, and need not be explained further herein; the method in each instance is the same, and follows approximately the process shown in FIG. 2.

The heat removal at high temperature, such as through conduits 47 in FIG. 3, can be used to provide a heat pump using this process. If the units main purpose is to be a heat pump, then line 29-30 may be extended so that point 30 is at the same pressure with point 26. Such system then may be similar to that shown in FIG. 5,

with the work from expander 79 fed back to system via the connecting shaft.

The apparatus of FIG. 1 and 3 were the subject of U.S. Pat. No. 3,972,194, by this applicant, in slightly different form. The apparatus of FIG. 4 and FIG. 5 are shown also in copending patent applications, Ser. Nos. 546,912 and 674,398, in slightly different form.

Other types of equipment can be used in addition to the types shown. Rotary piston, the wankel type rotary, and various gear types and other positive displacement type devices can be used. They may use the heat transfer fluid, or have jackets, or have heat exchangers. However, the compression process, as shown in FIG. 2, with constant volume heat addition, followed with heat addition either by using a compressor, or using external heat, is required, as disclosed in this invention.

In the drawings, the heat removal for external use, such as conduits 47 in FIG. 3, are shown as a part of the heat exchanger 41. A separate heat exchanger may be provided to contain lines 47 if desired.

In FIG. 4, the cylinder is shown with the cylinder head downward, and this is to provide better drainage of the heat transfer fluid. The cylinder may be set at any suitable orientation.

It should be also obvious that the working fluid being discharged by the compressor could be used directly as the heating fluid for the heat addition to the working fluid upstream of mechanical compression. Where the compressing unit is jacketed, or provided with suitable heat exchanger means this can be done. The same method, embodied in the work cycle of FIG. 2, still applies, even though the heat transfer fluid is not used. The heat transfer fluid used in the apparatus shown in the drawings, is merely a means for transferring the heat; other means for transferring the heat may be also used, without departing from the method shown in FIG. 2.

I claim:

1. A method of compressing a compressible working fluid comprising:

- confining said working fluid in an approximately constant volume and simultaneously adding heat into said working fluid increasing the temperature of said working fluid, and increasing the pressure of said working fluid;
- elevating the pressure of said working fluid by adding work into subsequently said working fluid in a mechanical compressing means;
- removing heat from said working fluid and transferring at least a portion of such heat into said working fluid upstream of said mechanical compressing means.

2. The method of claim 1 comprising the additional step of extracting work from said working fluid downstream of said heat removal.

3. The method of claim 1 comprising the additional step of removing heat from said working fluid downstream of said mechanical compressing means, and passing such heat out of the compressing system.

4. The method of claim 1 comprising the additional step of providing cooling for said working fluid during compression within said mechanical compressing means.

* * * * *

[54] **HEAT PUMP WITH TWO ROTORS**

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee,
Houston, Tex. 77042

[21] Appl. No.: **759,075**

[22] Filed: **Jan. 13, 1977**

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Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 407,665, Nov. 18, 1973, Pat. No. 4,047,392, Ser. No. 591,881, Jun. 30, 1975, abandoned, and Ser. No. 618,456, Oct. 1, 1975, Pat. No. 4,005,587.

[51] Int. Cl.² **F25B 3/00**

[52] U.S. Cl. **62/401; 62/501;**
165/88; 165/DIG. 12

[58] Field of Search 62/401, 499, 86, 501;
60/650, 682; 165/86, 88, DIG. 12

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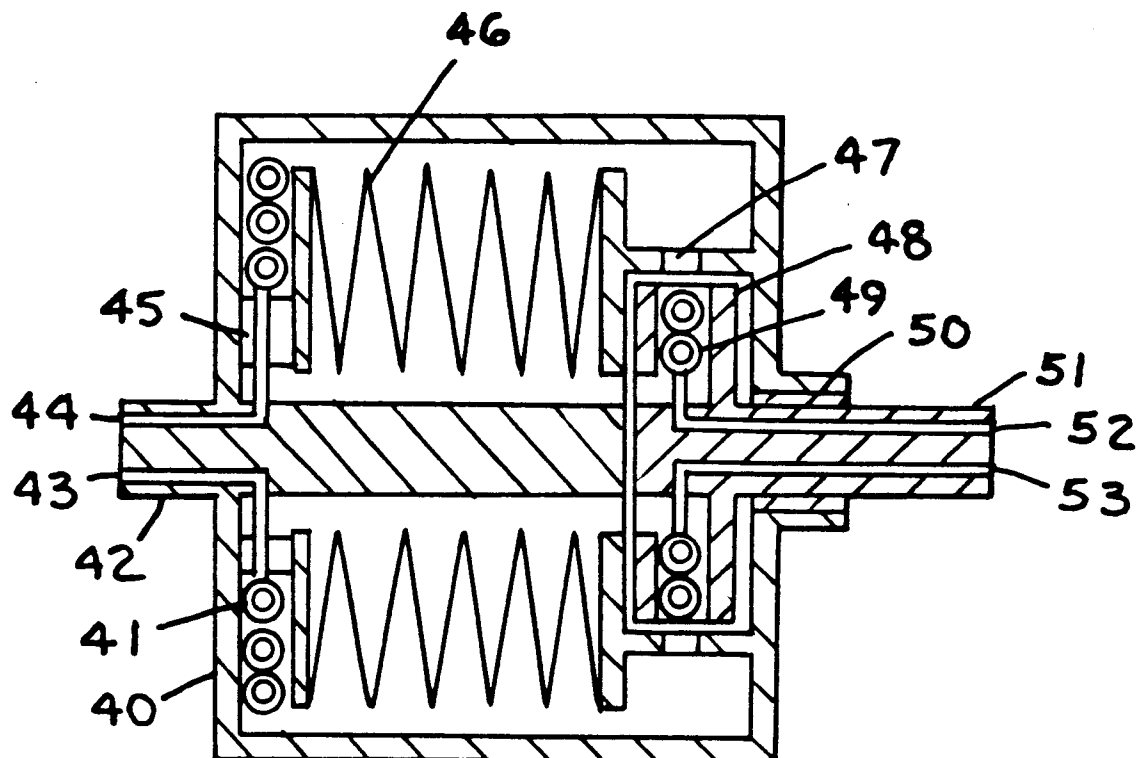
Primary Examiner—Charles J. Myhre

Assistant Examiner—Sheldon Richter

[57] **ABSTRACT**

A method and apparatus for generating heating and cooling by circulating a working fluid within passages carried by rotors, compressing said working fluid therewithin and removing heat from said working fluid in a heat removal heat exchanger and adding heat into said working fluid in a heat addition heat exchanger, all carried by said rotors. The working fluid is sealed within, and may be a suitable gas, such as nitrogen. A working fluid heat exchanger is also provided to exchange heat within rotor between two streams of said working fluid. In one arrangement, the unit uses two rotors, both rotating; in an alternate arrangement, one of the rotors may be held stationary. Applications include air conditioning service, and heating applications.

8 Claims, 3 Drawing Figures



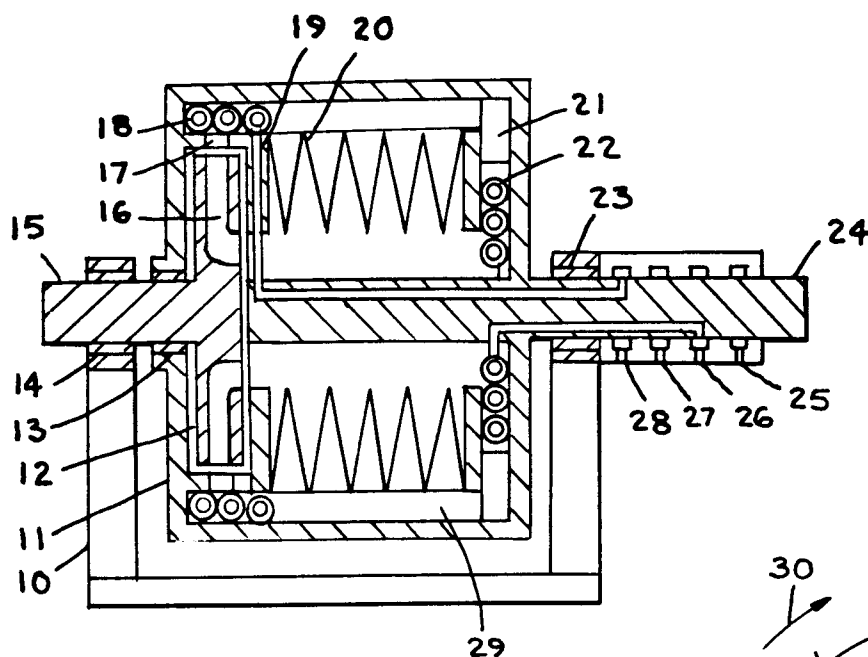


FIG. 1

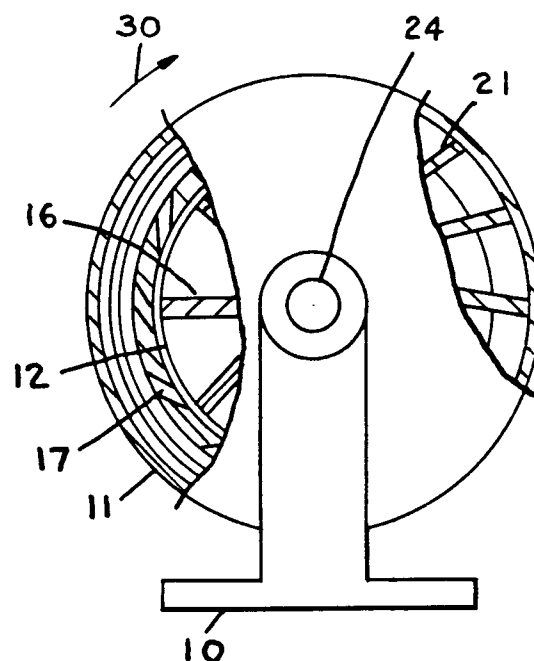


FIG. 2

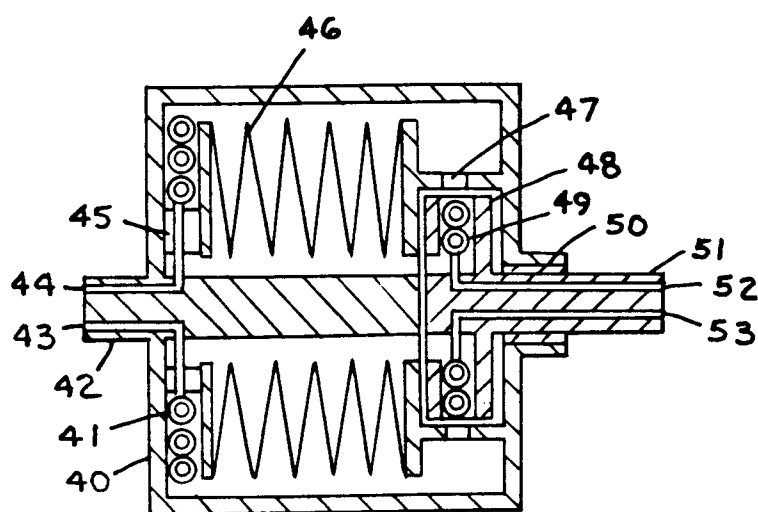


FIG. 3

HEAT PUMP WITH TWO ROTORS

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part application of "Dual Rotor Heat Exchanger" filed Nov. 18, 1973, Ser. No. 407,665, now U.S. Pat. No. 4,047,392. This application also is a continuation-in-part of "Heat Pump" filed June 30, 1975, Ser. No. 591,881, now abandoned. And this application also is a continuation-in-part of "Rotary Heat Exchanger with Cooling and Regeneration" filed Oct. 1, 1975, Ser. No. 618,456, now U.S. Pat. No. 4,005,587.

BACKGROUND OF THE INVENTION

This invention relates generally to devices for heat transfer from a lower temperature to a higher temperature by using a working fluid inclosed within a centrifuge rotor as an intermediate fluid to transport the heat.

Heat pumps have been known in the past but are complex and costly, and usually use a working fluid that is evaporated and condensed, which results in poor efficiency, and thus high energy cost.

SUMMARY OF THE INVENTION

It is an object of this invention to provide apparatus that is low in initial cost and has high thermal efficiency thus reducing cost of the power required to run it. It is further the object of this invention to provide a device and process wherein the losses that normally occur in bearings and seals, due to friction, are applied to the working fluid for its circulation, thus in effect eliminating the power loss due to such friction losses. Also, it is an object of this invention to provide the rotor with a working fluid heat exchanger to reduce needed rotor speeds.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of the device, and

FIG. 2 is an end view of the device.

FIG. 3 is an axial cross section of another form of the device.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, there is shown an axial cross section of the device. 10 is base, 11 is first rotor, 12 is second rotor, 13 is seal and bearing, 14 is bearing supporting shaft 15, 16 is fluid passage in second rotor, 17 is working fluid opening which may be a nozzle, 18 is first heat exchanger for heat removal from working fluid, 19 is first heat transfer fluid conduit, 20 is working fluid heat exchanger, in this instance formed from sheet metal like bellows, 21 are vanes, 22 is second heat exchanger for heat addition to working fluid, 23 is bearing supporting shaft 24, 25 and 26 are entry and exit for second heat transfer fluid, 27 and 28 are entry and exit for first heat transfer fluid, and 29 is a vane in peripheral passage.

In FIG. 2, an end view of the unit shown in FIG. 1 is illustrated. 10 is base, 11 is first rotor, 17 are fluid openings, 12 is second rotor, 16 are second rotor fluid passages with vanes, 30 indicates direction of rotation, 24 is first rotor shaft, and 21 are vanes.

In FIG. 3, the rotors are arranged differently, but perform the same functions, approximately, as in the unit of FIG. 1. 40 is first rotor, 41 is first heat exchanger

for heat removal from first fluid, 42 is first rotor shaft, 43 and 44 are entry and exit for first heat transfer fluid, 45 is conduit, 46 is working fluid heat exchanger, 47 are fluid openings which may be nozzles, 48 is second rotor, 49 is second heat exchanger for adding heat to the working fluid, 50 is bearing and seal, 51 is second rotor shaft, 52 and 53 are entry and exit for second heat transfer fluid.

In operation, the rotors are caused to rotate and the rotor cavities are filled with a suitable working fluid, which is usually a gas, such as nitrogen, air or other gaseous or vapor substance. Referring to FIG. 1, the second rotor rotates usually faster than the first rotor, and the working fluid is compressed by centrifugal force in passages 16, and also in the first rotor to some extent, after which heat is removed in heat exchanger 18, with such heat then being transported by the first heat transfer fluid out of the device. The working fluid then passes along the peripheral passage 29 and releases heat in heat exchanger 20, after which the fluid is expanded against centrifugal force in vanes 21 and in heat exchanger 22 where heat is added to the working fluid. After expansion, the working fluid passes along center passage and receives heat from heat heat exchanger 20, thus completing its work cycle.

The operation of the unit in FIG. 3 is similar, except that the second rotor usually rotates slower than the first rotor, and also, the second rotor may be kept stationary, if desired. Note that if the second rotor is held stationary, one may use dirty water as the second heat transfer fluid; normally, in rotating heat exchangers, the heat transfer fluid must be free of solids, which will collect in the heat exchanger due to centrifugal force, and block the heat exchanger, and by having a stationary heat exchanger, ordinary water may be used, such as water from a cooling tower.

In the unit of FIG. 1, the power input is normally to the second rotor, and the first rotor is allowed to rotate freely. In such usage, the rotor diameters are selected to provide, together with the friction loss in bearings, for the needed speed differential between the two rotors. With the second rotor rotating faster, necessary push for the working fluid is provided to keep the working fluid circulating. Alternately, the speed differential may be maintained by using a power transmission between the two rotors, such as a gearbox. In the unit of FIG. 3, the second rotor speed is slower than the speed of the first rotor, and where the rotor diameters are suitable, the second rotor may be held stationary, providing needed push for the working fluid for its circulation.

The working fluid heat exchanger 20 and 46, employ centrifugal force and varying gas density to obtain heat exchange between the two working fluid streams. Hot gas in the peripheral passage is lighter, and colder gas between the folds of the heat exchanger is colder, thus the cold gas is displaced by lighter gas by centrifugal force. Similarly, at the center passage, cold gas at center displaces hot gas between folds. Other types of heat exchangers may be used for the heat exchanger 20, including heat pipes, sheet metal discs, and finned tubing filled with a liquid.

The rotor may be encased within a vacuum tank, if desired, to reduce friction on rotor outer surfaces. The use of the working fluid heat exchanger 20 will reduce required rotor speeds to obtain required temperature differentials between the two heat transfer fluids, which then reduces friction losses on the rotor, which may eliminate the need for a vacuum tank.

Various modifications of this device may be made, and different types of heat exchangers used. Also, working fluid radial passages may be curved in various directions, one being the slope for vanes shown in FIG. 2, item 21. By using vane slopes and sloped passages, one can adjust the amount of work exchange between the working fluid and the rotor. Nozzles 47 are usually oriented to discharge backward, to generate some torque on the first rotor; similar nozzles may be also used in passages 21 of the unit of FIG. 1. Further, the heat exchanger 22, of FIG. 1, may be mounted on a stationary member, if desired, in manner shown in FIG. 3, and also, heat exchanger 18 may be mounted within rotor 12, if desired. The various components of the units may be interchanged, as desired.

I claim:

1. In a heat pump wherein a compressible working fluid is circulated radially outwardly in a
 - a. first fluid passage, said first passage contained in a first member, and radially inwardly toward center of rotation in a second fluid passage, said second passage contained in at least one of said first and second members, said first and second members coaxially arranged, at least one of said members being supported by a shaft for rotation;
 - said first and said second radial working fluid passages communicatingly connected at their respective outward ends by an outer passage and at their respective inward ends by an inner passage, said radial and outer and inner passages forming a closed loop extending at least partially through both of said members, a working fluid adapted to be circulating through said loop, means for com-

pressing said working fluid by centrifugal force within said loop with accompanying temperature increase, first heat exchange means for cooling said working fluid after compression, said first heat exchange means being carried by one of said members, a second heat exchange means, carried by one of said members, for regeneratively exchanging heat between said working fluid within said inner and outer passages, and a third heat exchange means carried by one of said members for heating said working fluid after said heat exchange between said working fluid within said inner and outer passages.

2. The heat pump of claim 1 wherein a first heat transfer fluid is circulated within said first heat exchange means to remove heat with said first heat exchange fluid entering and leaving via conduits near the center of rotation of said members.
3. The heat pump of claim 1 wherein a second heat transfer fluid is circulated within said third heat exchange means entering and leaving via conduits near the center of rotation of said members.
4. The heat pump of claim 1 wherein both of said members are rotors.
5. The heat pump of claim 4 wherein the two rotors rotate at different angular speeds.
6. The heat pump of claim 1 wherein at least one of said members is a rotor.
7. The heat pump of claim 6 wherein said second heat exchange means includes a plurality of folds.
8. The heat pump of claim 7 wherein said second heat exchange means is of bellows configuration.

* * * * *

[54] THERMODYNAMIC COMPRESSOR

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[21] Appl. No.: 708,863

[22] Filed: Jul. 26, 1976

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Primary Examiner—Charles J. Myhre

Assistant Examiner—Sheldon Richter

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 675,304, Apr. 9, 1976,
Pat. No. 4,060,989, and Ser. No. 600,312, Jul. 30, 1975,
Pat. No. 3,986,361.

[51] Int. Cl.² F25B 3/00

[52] U.S. Cl. 62/499; 165/DIG. 12

[58] Field of Search 165/86, 88, DIG. 12;
62/401, 402, 499, 501; 60/650, 682; 126/247;
415/1, 80, 116, 178

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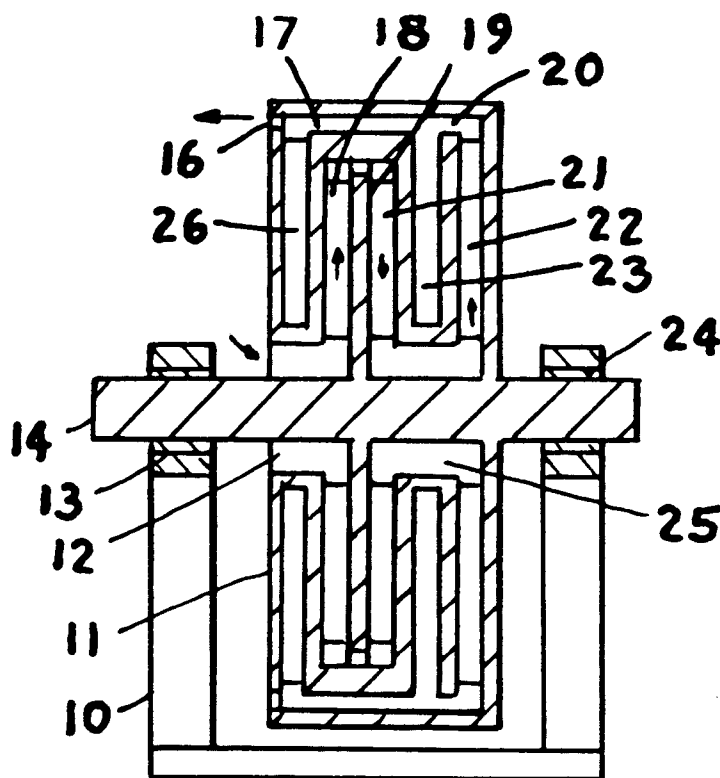
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[57] ABSTRACT

A method and apparatus for the thermodynamic assisted compression of gases wherein a gas is alternately compressed and expanded with addition of heat regeneratively. The basic apparatus and method are applicable to a variety of uses such as gas compression, turbines and in heat temperature boosters. Working fluids may be either gases or vapors. Heat may also be removed during compression steps and added during expansion steps. Process can be used with both steady flow and non-flow apparatus.

8 Claims, 6 Drawing Figures



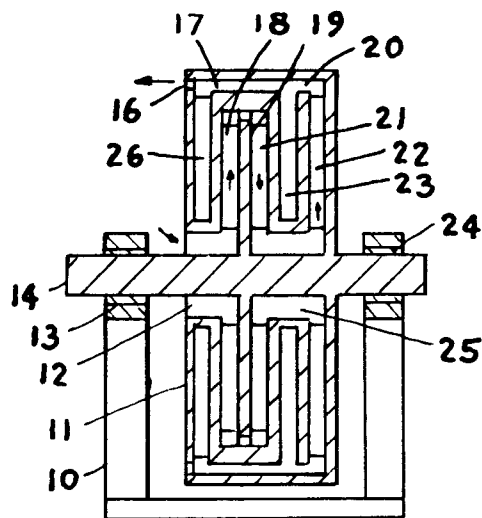


FIG. 1

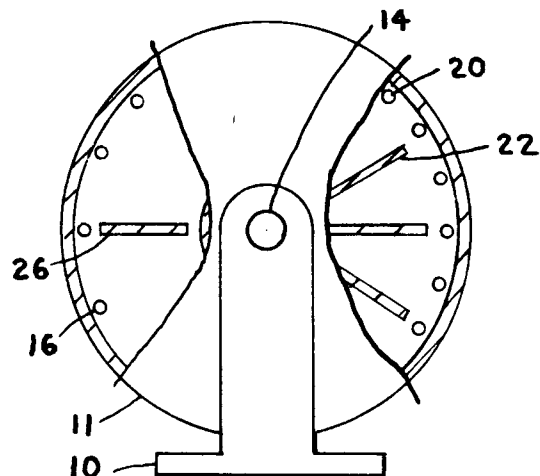


FIG. 2

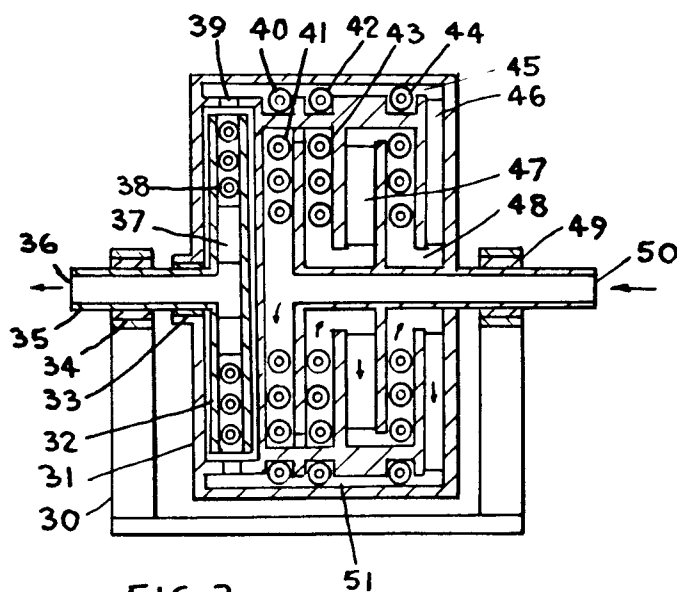


FIG. 3

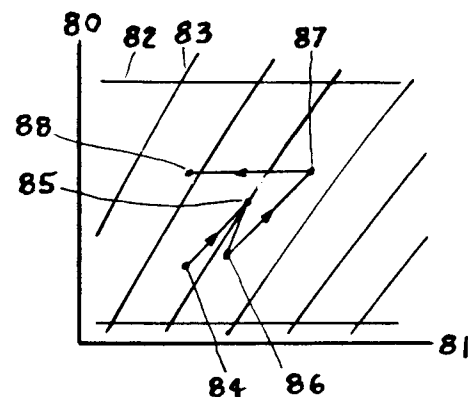


FIG. 5

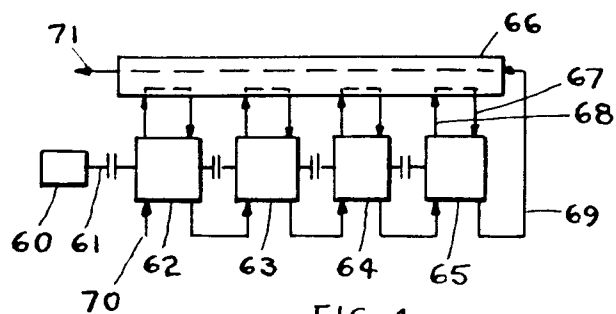


FIG. 4

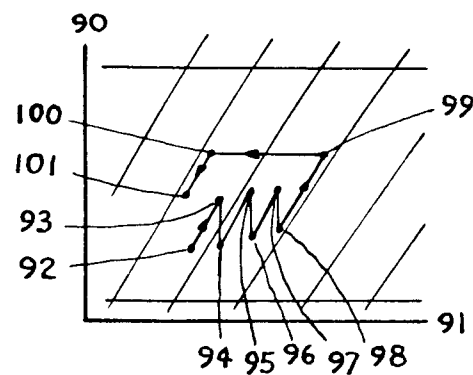


FIG. 6

THERMODYNAMIC COMPRESSOR

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation-in-part application of "Thermodynamic Machine", Ser. No. 675,304, filed 4-9-76, now U.S. Pat. No. 4,060,989, and a continuation-in-part of "Turbine with Regeneration," Ser. No. 600,312, filed 7-30-75, now U.S. Pat. No. 3,986,361.

BACKGROUND OF THE INVENTION

This invention relates generally to means to compress compressible fluids, such as compressible gases and vapors, wherein the fluid is alternately compressed and expanded, with the addition of heat during such compression and expansion steps. The Heat being added is usually obtained from the same working fluid regeneratively.

SUMMARY OF THE INVENTION

It is an object of this invention to provide a method and apparatus where the pressure of a working fluid is increased with reduced work input, by adding heat into the working fluid in a step type heat addition process, with alternate compression and expansion steps for the working fluid. The heat added into the working fluid is usually obtained by removing the heat from the heated and pressurized working fluid regeneratively. The compression and expansion steps are usually carried out in a non-flow type thermodynamic process; steady flow process may be also used with lesser efficiency.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the apparatus, and

FIG. 2 is an end view of the same unit.

FIG. 3 is a cross section of another form of the apparatus.

FIG. 4 is a fluid pressurization system using a similar process and using any type of apparatus.

FIG. 5 is a pressure-enthalpy or internal energy diagram for a typical working fluid with the work cycle for a pressure increaser shown thereon.

FIG. 6 is another pressure-enthalpy or internal energy diagram with a work cycle shown thereon.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, there is shown a centrifuge type rotor for the pressurization of compressible working fluids. 10 is base, 11 is rotor, 12 is fluid inlet, 13 and 24 are bearings supporting shaft 14, 26 is fluid space with optional vanes, 16 is fluid exit, 17 is fluid passage, 18 is a vane in working fluid outward extending passage, 19 is divider wall, 20 is fluid opening, 21 is vane in fluid inward extending passage, 22 is vane in outward extending fluid passage, 23 is fluid space with optional vane, 25 is fluid center passage.

In FIG. 2 an end view of the unit of FIG. 1 is shown. 10 is base, 11 is rotor, 16 are fluid exit openings, 26 is fluid space vane, 14 is shaft, 20 are fluid openings, 22 are vanes.

In FIG. 3, a cross section of another form of the apparatus is shown, with the regenerative heat exchangers being formed from finned tubing, and a means for extracting work from the compressed fluid before discharge by using a second rotor. Also, a means is pro-

vided for heat addition from external sources within the second rotor, if the unit is used as a turbine. 30 is base, 31 is first rotor, 32 is second rotor, 33, 34 and 49 are bearings, 35 is shaft, 36 is fluid exit, 37 is vane within second rotor, 38 is heating coil, 39 are fluid openings which are usually nozzles, 40 and 41 are parts of a regenerative heat exchanger, 42 and 43 are parts of another step of a regenerative heat exchanger, 44 is another regenerative heat exchanger, 45 is fluid opening, 46 and 47 are vanes in their fluid passages, 48 is fluid passage, 50 is fluid entry, 51 is fluid passage.

In FIG. 4, a system using the same work process shown for the centrifuge type apparatus, is illustrated. The heat is added into the working fluid regeneratively, and the fluid undergoes alternate expansion and compression, with heat addition. 60 is power unit, connected by shaft 61 into the apparatus, 70 is fluid entry, 69 is fluid conduit and 71 is working fluid exit, 62 and 64 are compressors and 63 and 65 are expanders, being heated by a heat transfer fluid being circulated from heat exchanger 66 via conduits such as 67 and 68. Usually, numerous steps of compression and expansion are required to obtain desired pressure gain, and the number of compression and expansion components may be as desired. Also, it should be noted that the heat can be transferred directly in the compression-expansion components from the working fluid into the working fluid stream being compressed, thus eliminating the heat transfer fluid.

In FIG. 5, a pressure-enthalpy or internal energy diagram is shown with a typical work cycle for the apparatus illustrated thereon. 80 is pressure line and 81 is enthalpy or internal energy line, 82 is constant pressure line and 83 is constant entropy line. The work cycle is 84-85-86-87 and 88, with the fluid entering at 84 and leaving at 88. The heat addition is shown to be carried out both during compression and expansion, but heat can be added both during compression only, and during expansion only, as desired. The work cycle shown is applicable to the apparatus of FIG. 1.

In FIG. 6, another pressure-internal energy or enthalpy diagram is shown. 90 is pressure line and 91 is enthalpy or internal energy line. The work cycle is 92-93-94-95-96-97-98-99-100-101. The working fluid enters at 92 and leaves at 101, which is shown for a typical compressor use. The line 100-101 can be extended to the same pressure as point 92, which would be for a typical turbine use; also, in such use, the lines 92 and 100-101 are usually isentropic, following the constant entropy line, and without heat addition or removal to reduce the cost of manufacture of the apparatus.

In operation, the rotor is caused to rotate, and the fluid enters the rotor via entry 12, and is compressed in outward extending passages formed by vanes 18 which usually also serve as heat exchange members, then the fluid is expanded in passages 21 with vanes serving to improve heat transfer, then the fluid is compressed again in passages 22 with the vanes improving heat transfer, and then the fluid passes along peripheral passages 17 to exit openings 16, and also serves as a heat source for the fluid being compressed via fluid spaces 23 and 26. The spaces 23 and 26 are open into the peripheral fluid passage 17 and allow the heated working fluid enter the spaces, thus serving as the heat source for the fluid being compressed. In spaces 23 and 26, centrifugal force together with density changes in the working fluid provide for fluid circulation.

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In FIG. 6, the heat is added into the working fluid during expansion, while during compression, for most of the steps, the process is with heat removal. The heat removal can be produced with apparatus such as is shown in FIG. 1, where heat may be removed during compression through wall 19 and passed directly to the expanding fluid in passage 21. Similarly, heat may be removed through the wall from passage 22 to to passage 23 and from there to passage 21. Where heat transfer is not desired, the walls can be thermally insulated. Similarly, the finned tubes of FIG. 3 may be so connected as to provide for heat removal during fluid compression, and heat exchange during expansion and compression steps.

The finned tubes of FIG. 3 are usually filled with a liquid fluid and connected in a closed loop to form a heat transfer means from the outer fluid passage into the fluid being compressed or expanded in the inner passages. Also, one may leave the heat exchangers out, as is done in passage 47 of FIG. 3, thus providing for approximately isentropic process. The heat exchanger 38 may be provided when the unit is used as a turbine with a heated heat transfer fluid supplied from external sources through shaft passages; also, the heat exchanger 38 may be provided in compressor applications if desired.

Applications for this device include as a fluid pressurizer, power generator, thrust generator, and generally as a component in any rotary or other device where it is desired to increase the pressure of the working fluid; one component type application obviously is in the compression stage of gas turbines, particularly for the type apparatus shown in FIG. 1.

The compression and expansion steps are shown to be approximately equal in magnitude in the pressure-internal energy diagrams. The steps do not need be equal; various amounts of compression and expansion may be applied as desired, and the amount of heat added and removed, per step, may be as desired.

The openings 16 of FIG. 1, may be nozzles. These nozzles may be directed to discharge the fluid in a desired direction, including axially and either forwardly or backwardly as may be required for the unit operation.

The system of FIG. 4 can also be used to increase the pressure of a fluid, and may be used as a component in systems such as a gas turbine. Thus, the method used in the rotary units of FIG. 1 and FIG. 3, may be also used by any type apparatus ordinarily used for compression and expansion of gases, and is particularly suited for non-flow positive displacement type equipment. In operation, the system of FIG. 4, the working fluid is alternately expanded and compressed, with heat addition usually during both compression and expansion, or heat

addition between the expansion and compression steps at constant volume. Heat removal may also be carried out during compression. The system of FIG. 4 may be either a non-flow or steady flow type as desired

I claim:

1. A compressor comprising:

a. a rotor mounted for rotation;

b. a continuous fluid passage within said rotor providing:

i. alternately outwardly and inwardly extending fluid passage portions for alternately compressing and expanding a working fluid flowing therein;

ii. an outwardly extending fluid passage means communicating downstream of the downward end of the alternately outwardly and inwardly extending fluid passage portion;

iii. a peripheral portion of the fluid passage communicating downstream of the outward end of the outwardly extending fluid passage means;

c. a heat exchange means carried by said rotor and providing for heat exchange between fluid entering and leaving said outwardly extending fluid passage means.

2. The compressor of claim 1 wherein said heat exchange means provides for heat addition into the working fluid flowing within said inwardly extending fluid passage portion.

3. The compressor of claim 1 wherein said heat exchange means provides for heat addition into the working fluid flowing within the outwardly extending fluid passage portion.

4. The compressor of claim 1 wherein an exit comprising a nozzle is provided for said working fluid from said rotor.

5. The thermodynamic compressor of claim 1 wherein said peripheral passage is approximately parallel with the rotor axis.

6. The thermodynamic compressor of claim 1 wherein said heat exchange means comprises a heat conductive rotor member to transfer heat from the fluid within the peripheral passage into the fluid within the inwardly extending fluid passage portion.

7. The thermodynamic compressing means of claim 1 wherein a rotating member (second rotor) with inwardly extending working fluid passages is provided downstream of said exit (exits) to receive said working fluid, with said rotating member being mounted for rotation.

8. The thermodynamic apparatus of claim 1 wherein a heat addition heat exchanger means is provided within said rotating member.

* * * * *

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United States Patent [19]**Eskeli**

[11]

4,124,993

[45]

Nov. 14, 1978[54] **REFRIGERATION MACHINE**[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee,
Houston, Tex. 77042[21] Appl. No.: **827,943**[22] Filed: **Aug. 26, 1977****Related U.S. Application Data**[63] Continuation-in-part of Ser. No. 595,389, Jul. 14, 1975,
Pat. No. 4,044,824, and a continuation-in-part of Ser.
No. 825,701, Aug. 18, 1977.[51] Int. Cl.² **F25B 9/00**[52] U.S. Cl. **62/86**[58] Field of Search 62/401, 499, 500, 501,
62/502, 504, 506, 507, 86, 88; 165/86, 88[56] **References Cited****U.S. PATENT DOCUMENTS**

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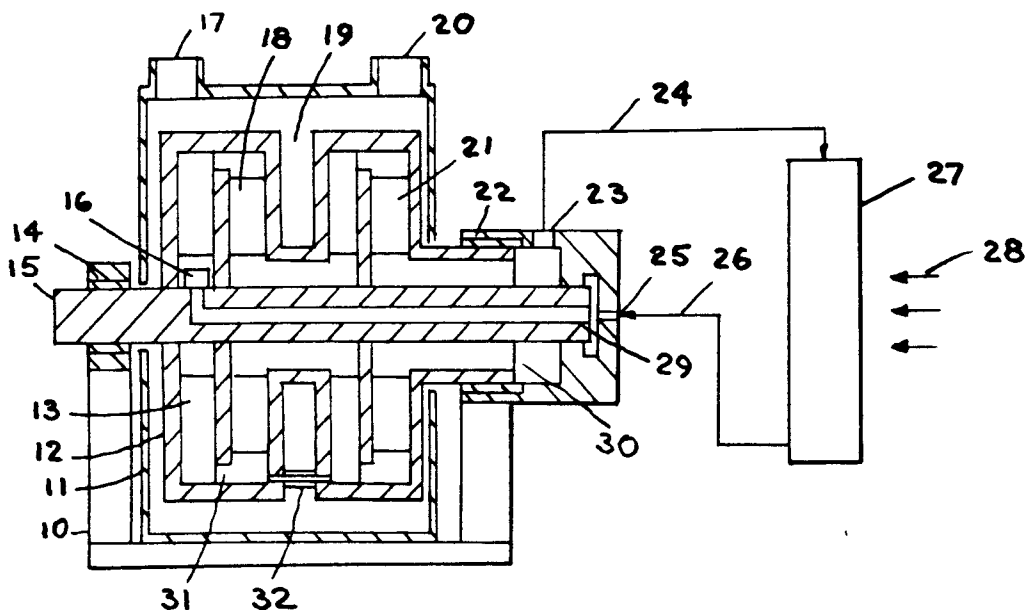
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Primary Examiner—Charles J. Myhre*Assistant Examiner*—Sheldon Richter

[57]

ABSTRACT

A method and apparatus for the generation of cooling or heating by using a centrifuge type heat exchanger wherein the refrigerant during heat addition undergoes a pressure increase. The apparatus where heat is added is a rotary heat exchanger with the refrigerant being within the heat exchanger and rotating with it, and the air which is releasing heat is outside of the heat exchanger in a stationary casing. Stationary or rotary heat exchanger may be used for heat rejection by the refrigerant. By reversing the air connections, one can also use the device for heating room air in air conditioning applications, while the air circulating around the heat exchanger is outside air. Refrigerant fluids for this device may be the usual fluids used for refrigeration, such as air, halogenated hydrocarbon, ammonia and others.

2 Claims, 4 Drawing Figures

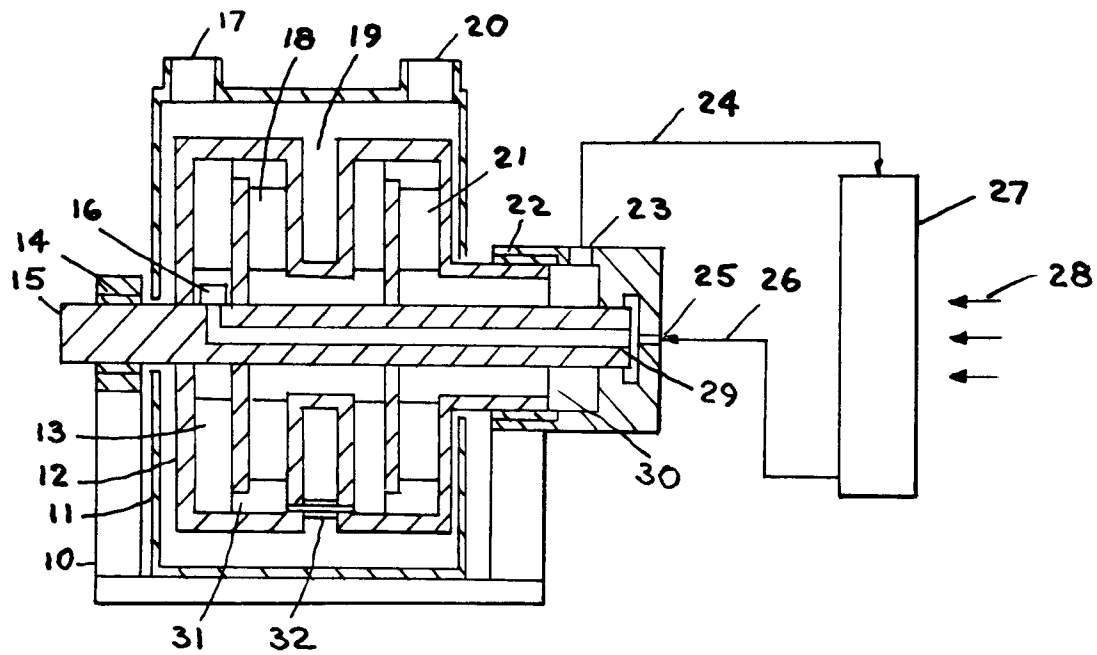


FIG. 1

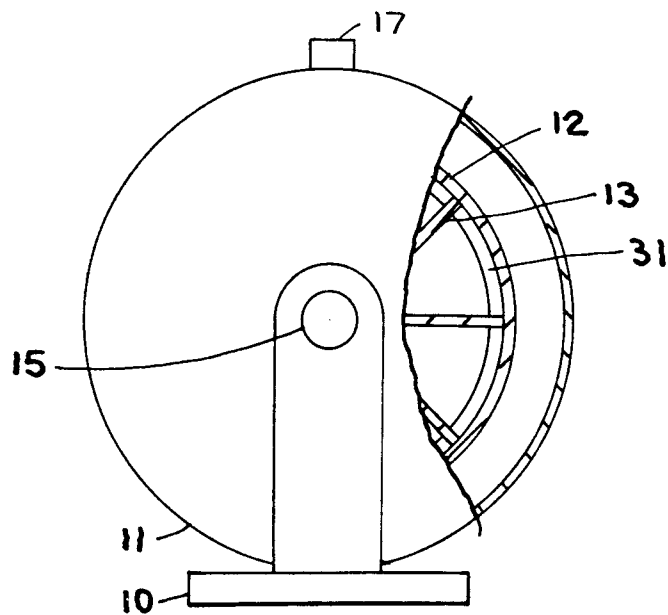


FIG. 2

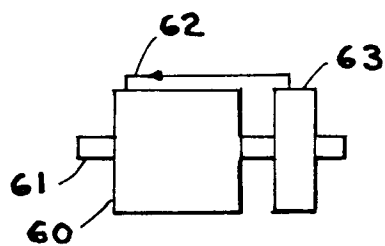


FIG. 4

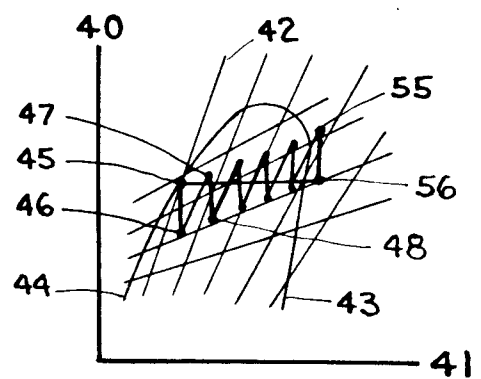


FIG. 3

REFRIGERATION MACHINE CROSS REFERENCES TO RELATED INVENTIONS

This is a continuation-in-part application of "Heat Exchangers", filed July 14, 1975, Ser. No. 595,389, now U.S. Pat. No. 4,044,824. And this is a continuation-in-part application of "Heat Exchanger", filed Aug. 18, 1977, Ser. No. 825,701.

BACKGROUND OF THE INVENTION

This invention relates generally to refrigeration machines and heat pumps wherein a refrigerant is circulated in a process where the fluid is condensed and vaporized, with heat being transported from a lower temperature to a higher temperature.

In previous machines of this type, the vaporization is usually done at a constant pressure in an evaporator coil, and condensing is also done in a constant pressure coil. These machines generally have a poor thermal efficiency and use large amounts of power.

SUMMARY OF THE INVENTION

It is an object of this invention to improve the thermal efficiency of the refrigeration or heating process to reduce the power consumption of the machine.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of the apparatus, with emphasis on the evaporation part of the machine.

FIG. 2 is an end view of the machine.

FIG. 3 is a pressure-enthalpy or internal energy diagram for the unit work cycle.

FIG. 4 is a schematic showing all parts on a single unit.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, therein is shown a cross section of the evaporation portions of the unit, and the condenser coil and connecting conduits are shown schematically. 10 is base, 11 is air shell, 12 is rotor, 13 is outward extending passage, 14 is shaft bearing, 15 is shaft, 16 is expansion device, usually a valve or a capillary tube, 17 and 20 are air outlet and inlet respectively, 18 is inwardly extending refrigerant passage, 19 is air space for transferring heat to the refrigerant fluid, 21 is inwardly extending passage, 22 is bearing and seal, 23 is gas discharge from rotary part, 24 is conduit to condenser coil, 27, 28 indicates air passing through the condenser coil, 25 is liquid inlet to shaft passage, 26 is liquid line, 29 is shaft liquid passage, 30 is gas exit from rotor, and 31 is peripheral passage, which may be a set of nozzles if desired.

In FIG. 2 an end view of the condensing portion is shown. 10 is base, 11 is air shell, 15 is shaft, 12 is rotor, 13 is vane in outwardly extending fluid passage, 31 is peripheral passage.

In FIG. 3, a pressure-enthalpy or internal energy diagram is shown for the refrigerant together with a work cycle. 40 is pressure line, 41 is enthalpy or internal energy line, 42 is entropy line. Expansion in an expansion valve is indicated by line 45-46, and the several compression-expansion steps with heat addition are shown by 46-47-48 to 55-56. The pressure at point 56 may be approximately the same as at point 45. Line 56-45 represents the constant pressure heat removal in the condenser coil; such heat removal may be at a vary-

ing pressure also, if desired. 44 and 43 are the saturated liquid and saturated vapor lines. Note that the gas is gaining in pressure during heat addition from point 46 to point 56; such pressure gain is usually sufficient without work addition to the refrigerant from the shaft; work can be added from shaft, if desired, or where the heat addition pressure gain is insufficient by itself.

In FIG. 4, a method is shown to provide for passing the liquid from the discharge of the condenser to the start of the evaporator, where the condenser is mounted on same shaft with the evaporator and rotates with it. Such arrangement makes possible to have the apparatus hermetic, without gas seals such as item 22 in FIG. 1. 60 is the evaporator portion of the rotor, 63 is the condensing portion of the rotor, the two usually mounted on a common shaft, 61 is shaft, and 62 is liquid line from the condenser to evaporator, passing usually within the rotors but shown here outside the rotors for clarity. The liquid is taken from the condenser space periphery, and passed to the evaporator portion periphery, with an expansion valve, or other expansion device to provide for the usual expansion service for the refrigerant. Shaft 61 is mounted on suitable bearings and base, and caused to rotate, and air is passed over the condensing coil in the usual manner.

In operation, the rotor is caused to rotate, and the refrigerant enters the outwardly and inwardly extending passages via liquid passage and valve 16. The liquid is evaporated by adding heat in the several compression-expansion steps, by adding heat from rotor exterior through heat conductive walls. Liquid will usually pass to rotor periphery, and the part that is not vaporized in the first compression-expansion step, can pass to the next step through veep holes and conduits such as item 32 in FIG. 1. The vapor gains in pressure within the several steps mainly due to heat addition during each expansion step. After the gas has been pressurized and heated and vaporized, it passes to the condenser where it is condensed by removing heat, and the liquid is then passed to the rotary evaporator portion, thus completing its cycle. The air to be cooled, is entering via entry 20 and passes in counterflow around the rotor, and exits then via exit 17, and is the refrigerated air. The air passing through the condenser is heated, and can serve as the heated air. Thus, the machine can be used as a heat pump by simply switching the air flows to the space to be conditioned, and outside air. The machine also circulates its refrigerant, as described.

The main advantage of this apparatus when compared with currently used standard refrigeration machines, is in reduced power consumption. The pressure gain for the fluid being heated, which is the refrigerant fluid, within the rotor, is nearly work free, with the pressure gain being a consequence of the heat addition. For work free operation, the outwardly and inwardly extending fluid passages within the rotor are made of same radial length, so that work added during each compression step is the same as the work recovered during each expansion step. Obviously such radial passages can be also of differing lengths, as desired, with resulting changes in the work quantities. Usually, numerous compression-expansion steps are used, to reduce the required rotor rotational speeds; two such steps are shown in FIG. 1. By using sufficient number of steps, a suitable pressure gain is obtained, and additional pressure can be provided by using some shaft work with suitably arranging the vanes in the radial passages by curving them and changing their radial length.

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The portion indicated by 63 in FIG. 4, may be made of similar construction as the evaporator portion, or it may be a fin and tube type rotor.

Applications of this device include the usual for air conditioning heating and refrigeration.

The sheet metal shell, item 11, may be attached to rotate with the rotor, if desired. Such rotation would reduce windage losses on the rotor to some extent, as the outer surface of the shell 11 is smooth.

The temperature of the working fluid is increased during the compression-expansion steps in the evaporating portion, generally from step to step. The liquid tubes, item 32, are made to allow the passage of unvaporized liquid, but are made small enough to prevent the passage of significant amounts of gas. The outward extending passages in the evaporator portion are made usually to be restricted for the tangential movement of the fluid, while the inwardly extending passages are wider in the tangential direction to allow some tangential movement and turbulence. The vanes in the radial passages may be also curved if desired.

Thermal insulation may be provided as desired. Such insulation may be desirable to prevent heat addition to the fluid during compression. Also, the stationary shell may be insulated.

The heat addition into the working fluid may be also from a liquid circulated in heat exchange relationship

with the working fluid in passages such as 18. The heat exchanger may then be finned tubing mounted within the rotor and supplied with a liquid heat transfer fluid from external sources via shaft passages. Usually, such liquid would pass in succession first through heat exchanger in passage 21 and then through heat exchanger in passage 18, with its temperature reduced as it passes from one step to next, in what is known as counterflow. Simultaneously, the working fluid would get warmer from step to step. Thus, this evaporation device can be used also to provide chilled water. Similarly, the condenser heat exchanger can be provided with a liquid heat transfer fluid; thus, heated water can be produced by the machine.

I claim:

1. A thermodynamic method comprising:

- a. passing a fluid through a pressure reducing expansion valve;
- b. subsequently vaporizing said fluid in a continuously rotating evaporator-compressor comprising alternate compression-expansion steps with simultaneous addition of heat and accompanying pressure increase.

2. The thermodynamic method of claim 1 wherein said fluid subsequent to the vaporizing is condensed by removing heat from said fluid.

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[54] METHOD OF FLUID PRESSURIZATION

[76] Inventor: Michael Eskeli, 7994-41 Locke Lee, Houston, Tex. 77042

[21] Appl. No.: 825,701

[22] Filed: Sep. 6, 1977

[51] Int. Cl.² F04B 41/00

[52] U.S. Cl. 417/53; 60/650; 62/86; 417/243

[58] Field of Search 60/650, 682; 62/86, 62/87, 88, 401, 402, 499, 500; 417/53, 243, 247; 165/88, 90, DIG. 12, 86

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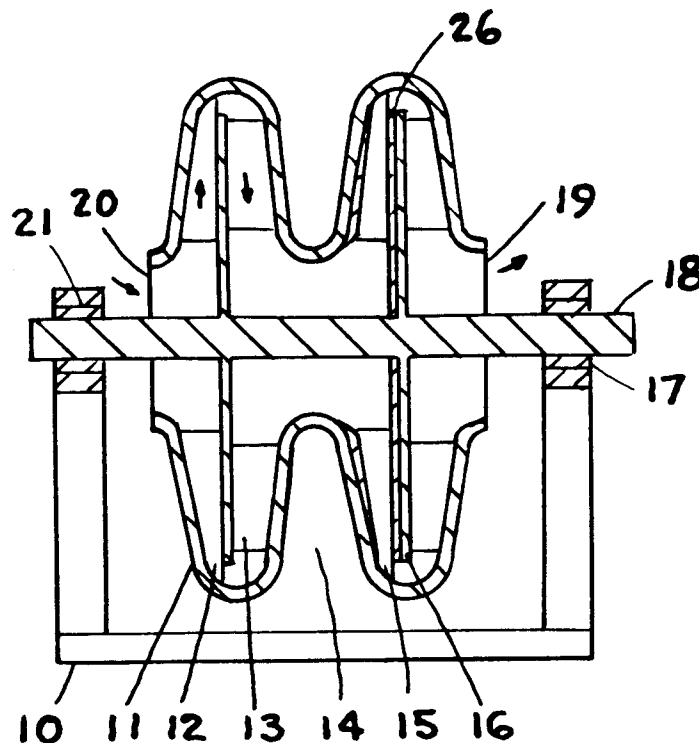
26184	11/1907	United Kingdom	165/88
936059	9/1963	United Kingdom	165/90

Primary Examiner—Sheldon Jay Richter

[57] ABSTRACT

A method and apparatus for transferring heat regeneratively in a rotating heat exchanger. The heat exchanger provides for the cold gas stream to be heated alternate compression and expansion with heat addition usually during the expansion phase of the compression-expansion step from a warmer fluid. The fluid to be heated is usually a gas, while the fluid releasing heat may be either a gas or liquid. The heat element is a folded member usually in the form of bellows mounted concentric with rotor shaft, with baffles within to provide for outward and inward movement of the gas. Due to the heat addition especially during the expansion steps, the gas to be heated gains in pressure and usually this pressure gain is work free, and in magnitude is similar to the constant volume heat addition pressure gain which is well known.

2 Claims, 3 Drawing Figures



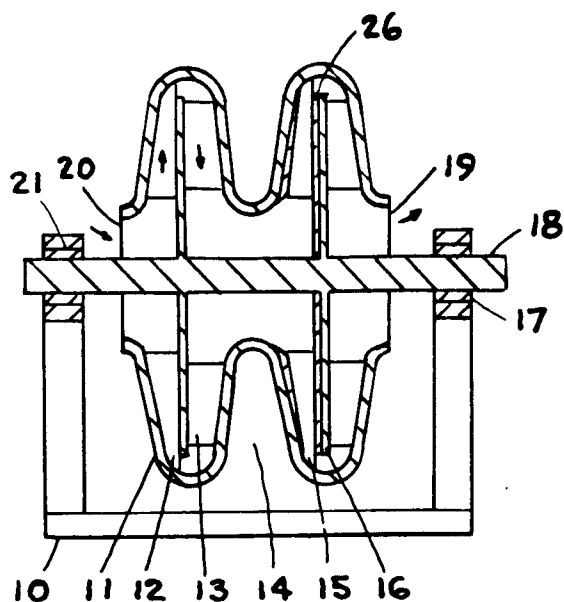


FIG. 1

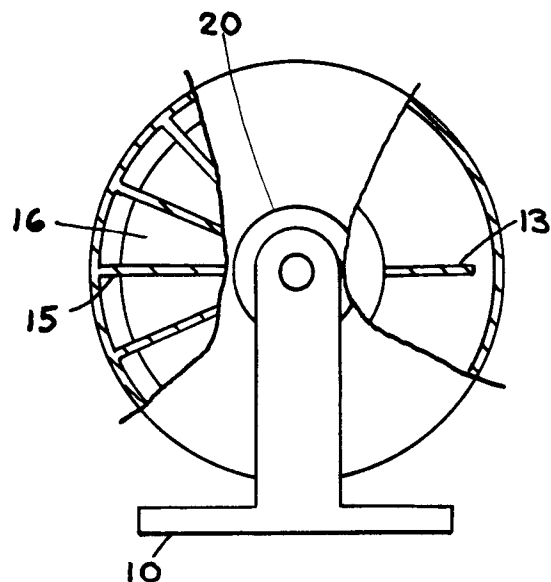


FIG. 2

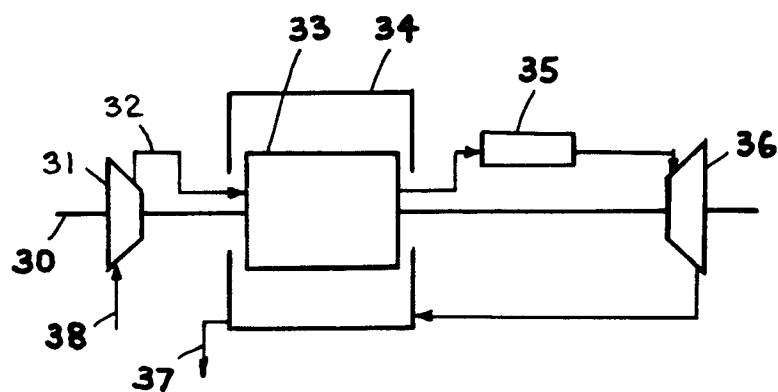


FIG. 3

METHOD OF FLUID PRESSURIZATION

BACKGROUND OF THE INVENTION

This invention relates generally to heat exchangers where a colder fluid stream is heated by a warmer fluid stream, and particularly to rotating heat exchangers where a gaseous fluid is alternately compressed and expanded with heat addition to provide for a pressure gain for the fluid to be heated.

In my previous patent application "Heat Exchanger", I disclosed a device which was provided with a means for alternately compressing and expanding a fluid to be heated within a bellows type heat exchange member, and an attached shell containing the fluid releasing heat, with the entire apparatus rotating.

SUMMARY OF THE INVENTION

It is the object of this invention to provide a rotary heat exchanger where a the fluid to be heated is within the heat exchanger and rotates with it, and where the fluid releasing heat is within a stationary housing, or alternately no housing is provided for the fluid releasing heat.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial cross section of the heat exchanger, and FIG. 2 is an end view of the same unit.

FIG. 3 is a schematic illustration of one application for the device of this invention, relating to a gas turbine.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, therein is shown an axial cross section of the heat exchanger. 10 is base, 11 is rotating heat exchange member supported by shaft 18 and bearings 17 and 21. Fluid to be heated enters via entry 20, passes outwardly to periphery via passage defined by vanes 12, returns back to center via passages defined by vanes 13, passes out again via passages 15 and inwardly via passages to exit 19. 16 is divider wall, 26 indicates optional thermal insulation layer, and 14 is space for the fluid releasing heat which is outside of the rotating member.

In FIG. 2, an end view is shown. 10 is base, 15 is a vane, 16 is dividing wall, 20 is fluid entry, and 13 is a vane.

In FIG. 3, an application of the heat exchanger is shown, adapted for use as a regenerator in a gas turbine. 30 is shaft, 31 is compressor, 36 is turbine, 35 is burner. Air enters via 38, passes after compression via 32 to center of regenerative heat exchanger 33, and thence to burner 35 and after heating to turbine 36, and from there the exhaust gases pass to regenerator housing 34, and after heat removal, are exhausted, via 37.

In operation, the rotor is caused to rotate, and the gas to be heated enters via entry 20, is alternately compressed and expanded with heat addition from the fluid outside the rotor, and after heating and a gain in pressure, the gas leaves via exit 19. alternately, the passages such as 13 may be made slightly longer than passages such as 12, and the heat exchanger will generate some power; in such case, the pressure gain will be less. Additionally, various other methods may be used to extract work, rather than to have pressure gain, within the heat exchanger for the gas to be heated.

It should be noted that in usual practice, the radial length of passages such as 12 and 13, are not great, and

thus the pressure gain per each step is also small. By using numerous compression-expansion steps, higher gains in pressure may be obtained, in aggregate.

The main advantage of the use of the stationary housing such as 34 in FIG. 3, is that dirt and soot as may be produced during combustion, will have a lesser tendency to accumulate within the housing 34, which is stationary. In a unit where the housing 34 rotates with the rotor, dirt and ash would be separated and collected within such housing, causing a maintenance problem.

In a gas turbine as shown in FIG. 3, the pressure gain in the heat exchanger 33 may be sufficient for the turbine, thus allowing the elimination of the separate compressor 31. In such applications, the thermal efficiency of the gas turbine is greatly improved.

The heat exchange member is usually made of sheet-metal stampings, with heat exchange directly from one fluid to another through the member walls. Other type of heat exchange may be used such as heat pipes crossing the heat exchange member wall for increased heat exchange area. Fins also may be provided to enhance heat transfer.

Generally, for lower rotor speeds, numerous compression-expansion steps are required, while for high rotor speeds, fewer steps are necessary. Fluid velocity through passages such as 12 extending outwardly, is usually higher than the fluid velocity through passage 13 extending inwardly. Also, number of vanes in passage 12 is usually greater for closer fluid tangential movement control, than in passage 13, where tangential movement may be allowed for the fluid passing through. The difference in the number of vanes is indicated in FIG. 2, for the two passages.

Thermal insulation may be provided in the outwardly extending fluid passages if desired; usually the same effect may be provided by having a higher radial velocity for the fluid in the outwardly extending passages than in other areas of the heat exchanger.

Note that the fluid within this heat exchanger may be either cooled or be heated by the fluid outside, depending of temperatures of the two fluids. Where the fluid within the heat exchanger is cooled, the main heat exchange occurs during compression, that is, during the travel of the fluid outwardly within the heat exchanger member folds, and for heating the main heat exchange occurs during expansion as the fluid travels inwardly within the heat exchanger member folds.

The heat exchange member is usually made in the form of folds similar to bellows, from sheetmetal. The vanes then can be formed to the sheetmetal, and the entire assembly supported by the rotor shaft through baffles, such as 16 in FIG. 1. Spaces such as 14 are left between the folds to allow for circulation of the fluid outside for heat transfer. The sheetmetal walls of the folds may also be provided with grooves for enhanced heat transfer, in areas where heat transfer is desired.

To provide for starting of fluid flow through the heat exchanger, the entry pressure at opening 20 should be higher than at exit 19. Alternately, the diameter of exit 19 may be made larger than entry 20.

For the several compression-expansion steps, the temperature of the fluid to be heated increases from step-to-step. The fluid to be heated and the fluid outside of the heat exchanger are usually in counterflow as shown in FIG. 3.

I claim:

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1. A method of heat exchange and fluid pressurization comprising:

- a. alternately compressing and expanding a compressible fluid in a plurality of compression-expansion steps and simultaneously adding heat into said fluid during the passage of said fluid through the compression-expansion steps;
- b. causing the temperature of said fluid to increase from step-to-step during the passage of the fluid

within the plurality of said compression-expansion steps.

2. The method of claim 1 wherein the amount of work added into the fluid and the work extracted from the fluid is approximately the same in a compression-expansion step and wherein there is a pressure gain for said fluid passing through said compression-expansion step due to heat addition.

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[54] **THERMODYNAMIC COMPRESSOR METHOD**

[76] Inventor: **Michael Eskeli**, 7994-41 Locke Lee, Houston, Tex. 77042

[21] Appl. No.: **884,832**

[22] Filed: **Mar. 9, 1978**

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Primary Examiner—Sheldon Jay Richter

Related U.S. Application Data

[62] Division of Ser. No. 708,863, Jul. 26, 1976, Pat. No. 4,107,945.

[51] **Int. Cl.²** **F25B 3/00**

[52] **U.S. Cl.** **62/88; 62/499; 62/501; 165/88; 165/DIG. 12**

[58] **Field of Search** **62/86-88, 62/401, 499, 501; 60/650, 682; 415/178; 165/88, 90, DIG. 12; 417/53, 243, 247**

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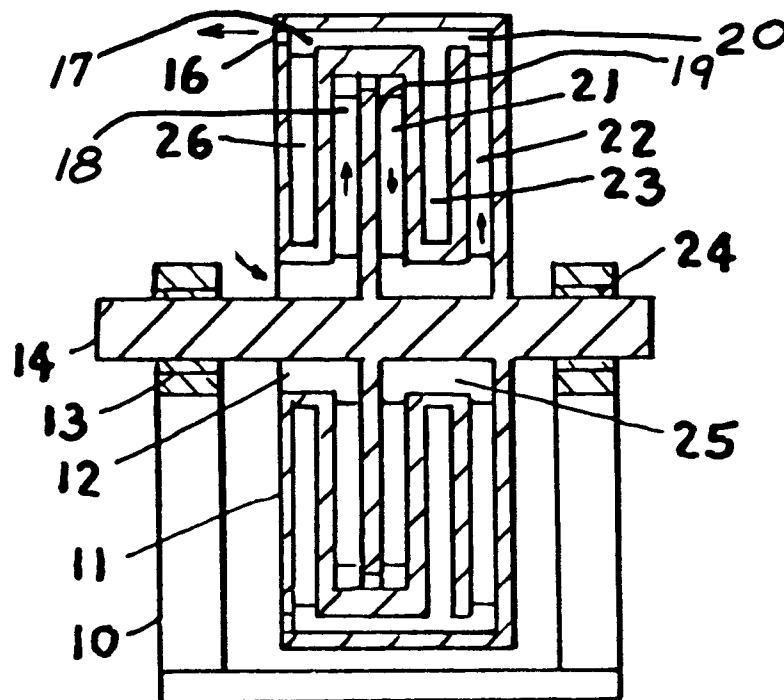
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[57] **ABSTRACT**

A method for the thermodynamic assisted compression of gases wherein a gas is alternately compressed and expanded with addition of heat regeneratively. The basic method is applicable to a variety of uses such as gas compression, turbines and in heat temperature boosters. Working fluids may be either gases or vapors. Heat may also be removed during compression steps and added during expansion steps. Process can be used with both steady flow and non-flow apparatus.

8 Claims, 6 Drawing Figures



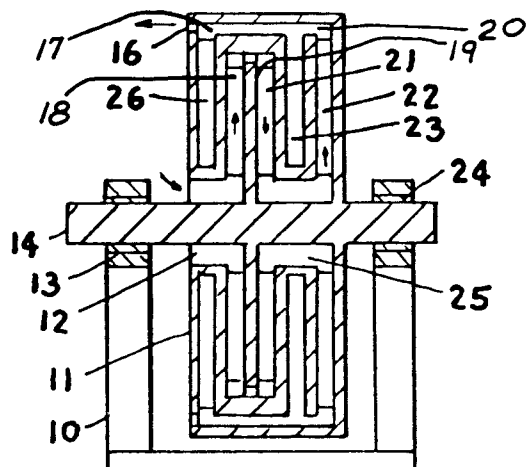


FIG. 1

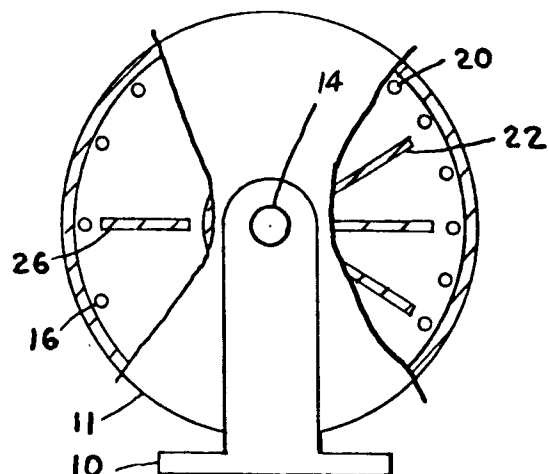


FIG. 2

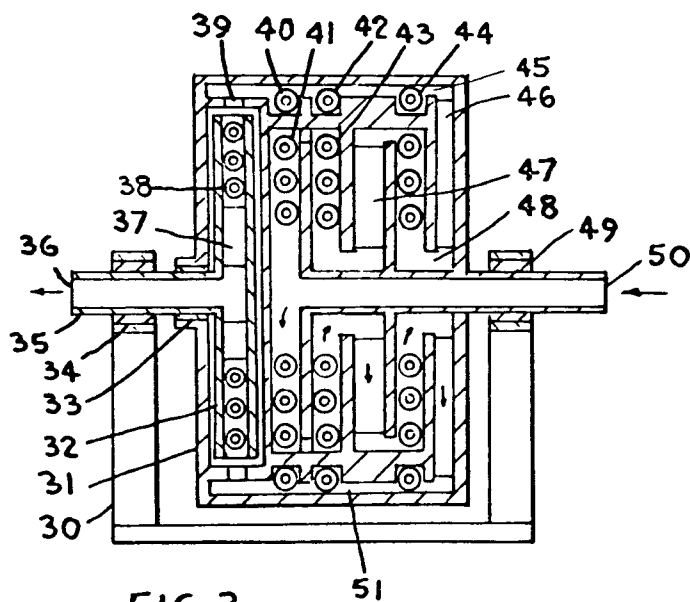


FIG. 3

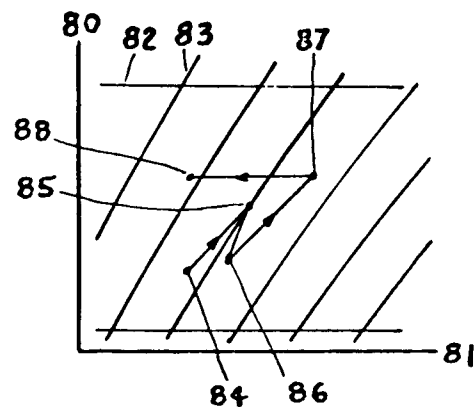


FIG. 5

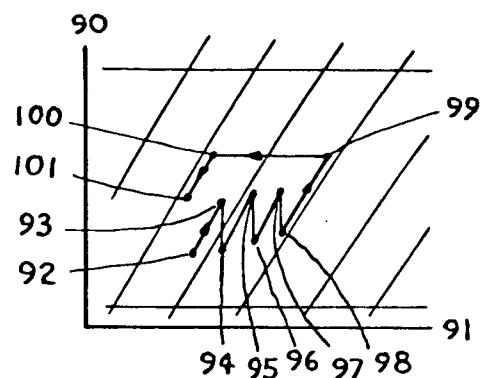


FIG. 6

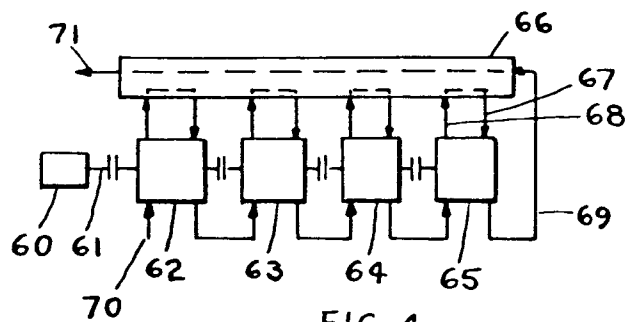


FIG. 4

THERMODYNAMIC COMPRESSOR METHOD

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a divisional application of "Thermodynamic Compressor", filed 7/26/76, Ser. No. 708,863, now U.S. Pat. No. 4,107,945.

BACKGROUND OF THE INVENTION

This invention relates generally to means to compress compressible fluids, such as compressible gases and vapors, wherein the fluid is alternately compressed and expanded, with the addition of heat during such compression and expansion steps. The Heat being added is usually obtained from the same working fluid regeneratively.

SUMMARY OF THE INVENTION

It is an object of this invention to provide a method and apparatus where the pressure of a working fluid is increased with reduced work input, by adding heat into the working fluid in a step type heat addition process, with alternate compression and expansion steps for the working fluid. The heat added into the working fluid is usually obtained by removing the heat from the heated and pressurized working fluid regeneratively. The compression and expansion steps are usually carried out in a non-flow type thermodynamic process; steady flow process may be also used with lesser efficiency.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross section of one form of the apparatus, and

FIG. 2 is an end view of the same unit.

FIG. 3 is a cross section of another form of the apparatus.

FIG. 4 is a fluid pressurization system using a similar process and using any type of apparatus.

FIG. 5 is a pressure-enthalpy or internal energy diagram for a typical working fluid with the work cycle for a pressure increaser shown thereon.

FIG. 6 is another pressure-enthalpy or internal energy diagram with a work cycle shown thereon.

DESCRIPTION OF PREFERRED EMBODIMENTS

Referring to FIG. 1, there is shown a centrifuge type rotor for the pressurization of compressible working fluids. 10 is base, 11 is rotor, 12 is fluid inlet, 13 and 24 are bearings supporting shaft 14, 26 is fluid space with optional vanes, 16 is fluid exit, 17 is fluid passage, 18 is a vane in working fluid outward extending passage, 19 is divider wall, 20 is fluid opening, 21 is vane in fluid inward extending passage, 22 is vane in outward extending fluid passage, 23 is fluid space with optional vane, 25 is fluid center passage.

In FIG. 2 an end view of the unit of FIG. 1 is shown. 10 is base, 11 is rotor, 16 are fluid exit openings, 26 is fluid space vane, 14 is shaft, 20 are fluid openings, 22 are vanes.

In FIG. 3, a cross section of another form of the apparatus is shown, with the regenerative heat exchangers being formed from finned tubing, and a means for extracting work from the compressed fluid before discharge by using a second rotor. Also, a means is provided for heat addition from external sources within the second rotor, if the unit is used as a turbine. 30 is base,

31 is first rotor, 32 is second rotor, 33, 34 and 49 are bearings, 35 is shaft, 36 is fluid exit, 37 is vane within second rotor, 38 is heating coil, 39 are fluid openings which are usually nozzles, 40 and 41 are parts of a regenerative heat exchanger, 42 and 43 are parts of another step of a regenerative heat exchanger, 44 is another regenerative heat exchanger, 45 is fluid opening, 46 and 47 are vanes in their fluid passages, 48 is fluid passage, 50 is fluid entry, 51 is fluid passage.

In FIG. 4, a system using the same work process shown for the centrifuge type apparatus, is illustrated. The heat is added into the working fluid regeneratively, and the fluid undergoes alternate expansion and compression, with heat addition. 60 is power unit, connected by shaft 61 into the apparatus, 70 is fluid entry, 69 is fluid conduit and 71 is working fluid exit, 62 and 64 are compressors and 63 and 65 are expanders, being heated by a heat transfer fluid being circulated from heat exchanger 66 via conduits such as 67 and 68. Usually, numerous steps of compression and expansion are required to obtain desired pressure gain, and the number of compression and expansion components may be as desired. Also, it should be noted that the heat can be transferred directly in the compression-expansion components from the working fluid into the working fluid stream being compressed, thus eliminating the heat transfer fluid.

In FIG. 5, a pressure-enthalpy or internal energy diagram is shown with a typical work cycle for the apparatus illustrated thereon. 80 is pressure line and 81 is enthalpy or internal energy line, 82 is constant pressure line and 83 is constant entropy line. The work cycle is 84-85-86-87 and 88, with the fluid entering at 84 and leaving at 88. The heat addition is shown to be carried out both during compression and expansion, but heat can be added both during compression only, and during expansion only, as desired. The work cycle shown is applicable to the apparatus of FIG. 1.

In FIG. 6, another pressure-internal energy or enthalpy diagram is shown. 90 is pressure line and 91 is enthalpy or internal energy line. The work cycle is 92-93-94-95-96-97-98-99-100-101. The working fluid enters at 92 and leaves at 101, which is shown for a typical compressor use. The line 100-101 can be extended to the same pressure as point 92, which would be for a typical turbine use; also, in such use, the lines 92 and 100-101 are usually isentropic, following the constant entropy line, and without heat addition or removal to reduce the cost of manufacture of the apparatus.

In operation, the rotor is caused to rotate, and the fluid enters the rotor via entry 12, and is compressed in outward extending passages formed by vanes 18 which usually also serve as heat exchange members, then the fluid is expanded in passages 21 with vanes serving to improve heat transfer, then the fluid is compressed again in passages 22 with the vanes improving heat transfer, and then the fluid passes along peripheral passages 17 to exit openings 16, and also serves as a heat source for the fluid being compressed via fluid spaces 23 and 26. The spaces 23 and 26 are open into the peripheral fluid passage 17 and allow the heated working fluid enter the spaces, thus serving as the heat source for the fluid being compressed. In spaces 23 and 26, centrifugal force together with density changes in the working fluid provide for fluid circulation.

In FIG. 6, the heat is added into the working fluid during expansion, while during compression, for most

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of the steps, the process is with heat removal. The heat removal can be produced with apparatus such as is shown in FIG. 1, where heat may be removed during compression through wall 19 and passed directly to the expanding fluid in passage 21. Similarly, heat may be removed through the wall from passage 22 to to passage 23 and from there to passage 21. Where heat transfer is not desired, the walls can be thermally insulated. Similarly, the finned tubes of FIG. 3 may be so connected as to provide for heat removal during fluid compression, and heat exchange during expansion and compression steps.

The finned tubes of FIG. 3 are usually filled with a liquid fluid and connected in a closed loop to form a heat transfer means from the outer fluid passage into the fluid being compressed or expanded in the inner passages. Also, one may leave the heat exchangers out, as is done in passage 47 of FIG. 3, thus providing for approximately isentropic process. The heat exchanger 38 may be provided when the unit is used as a turbine with a heated heat transfer fluid supplied from external sources through shaft passages; also, the heat exchanger 38 may be provided in compressor applications if desired.

Applications for this device include as a fluid pressurizer, power generator, thrust generator, and generally as a component in any rotary or other device where it is desired to increase the pressure of the working fluid; one component type application obviously is in the compression stage of gas turbines, particularly for the type apparatus shown in FIG. 1.

The compression and expansion steps are shown to be approximately equal in magnitude in the pressure-internal energy diagrams. The steps do not need be equal; various amounts of compression and expansion may be applied as desired, and the amount of heat added and removed, per step, may be as desired.

The openings 16 of FIG. 1, may be nozzles. These nozzles may be directed to discharge the fluid in a desired direction, including axially and either forwardly or backwardly as may be required for the unit operation.

The system of FIG. 4 can also be used to increase the pressure of a fluid, and may be used as a component in systems such as a gas turbine. Thus, the method used in the rotary units of FIG. 1 and FIG. 3, may be also used by any type apparatus ordinarily used for compression

and expansion of gases, and is particularly suited for non-flow positive displacement type equipment. In operation, the system of FIG. 4, the working fluid is alternately expanded and compressed, with heat addition usually during both compression and expansion, or heat addition between the expansion and compression steps at constant volume. Heat removal may also be carried out during compression. The system of FIG. 4 may be either a non-flow or steady flow type as desired.

I claim:

1. A method of compressing a compressible fluid comprising:

- a. alternately compressing and expanding said fluid in a first compressing and expanding means; subsequently
- b. compressing said fluid in a second compressing means with accompanying temperature increase;
- c. removing heat from said fluid downstream of the second compressing means and adding said heat into said fluid during the alternate compressing and expanding steps, within either said first compressing or expanding means.

2. The method of claim 1 wherein the heat addition into said fluid during the alternate first compression or (and) expansion is within said expanding means.

3. The method of claim 1 wherein the heat removal from said fluid downstream of the second compressing means is at approximately constant pressure.

4. The method of claim 3 wherein said heat removal at approximately constant pressure is performed within a centrifuge type machine.

5. The method of claim 1 wherein a separate heat transfer fluid is provided for removing the heat from the fluid downstream of the second compressing means and adding said heat into the fluid during the first alternate compressing or expanding means.

6. The method of claim 1 wherein the heat addition into said fluid during the alternate compression and expansion is within said first compressing means.

7. The method of claim 1 wherein said fluid is further expanded in another expansion means downstream of the heat removal and second compressing steps.

8. The method of claim 1 wherein in the first alternate compressio or expansion steps the work input during the compression and the work output during the expansion by the said fluid are approximately equal.

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Free Energy Update

Eskeli Thermodynamic Engine

In April of this year, Paul Carlson made us aware of a local inventor, Mr. Michael Eskeli, who had developed two types of engines which take advantage of the temperature differential utilized in Thermodynamics to develop a motive force.

Mr. Eskeli became incensed at an article written in the Dallas Times Herald concerning the failure of science to come up with alternative power systems not dependent on petroleum technologies. His concern prompted a letter to the editor on April 24 89'.

"I have had the patents for fuelless power generators, work-free heat pumps and other related items, 56 of them issued in the mid-70s, and am still waiting to collect a first nickel for labors and expense. And the indications are that these machines will not be in use before 2110 or so."

"The same probably will be true for the fusion people, excited by their discoveries now. There simply is too much oil still around to allow competition."

Based on this letter, we contacted Mr. Eskeli and found him to be authentic in his statement that he did indeed have numerous patents on fuelless engines and related items.

We consequently asked him to speak to a meeting of the Vanguard group to explain his principles of operation. Several weeks later, a demonstration was arranged to actually witness the device in action. Mr. Eskeli agreed to allow a very small group to meet with him in his home and agreed to demonstrate his device to a single person. Pictures of his disassembled Gas Turbine engine are included in this newsletter to show the principle of operation.

The meeting took place and Eskeli allowed two of us to witness the engine in his workshop the next day. We found a device with an approximate diameter of 4 feet and a length of 5 feet which was attached to a huge electrical motor by a belt. Michael explained that he had a small problem which did not allow the unit to operate under its own power, so he simply powered it from an electric motor to let us see how it worked. Subsequent communications have been unrewarding in the hope of a "real-time" demonstration. Until a working model is produced, we have decided to press on to other technologies and systems...JWD